Technical Report SL-94-15
July 1994

AD-A283 088

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US Army Corps
of Engineers
Waterways Experiment
Station

A Model for Assessment of Dynamic Interaction Between Magnetically Levitated Vehicles and Their Supporting Guideways

by James C. Ray, Mostafiz R. Chowdhury

94-25150

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Prepared for Headquarters, U.S. Army Corps of Engineers
National Maglev Initiative
and U.S. Army Engineer Division, Huntsville

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Final report

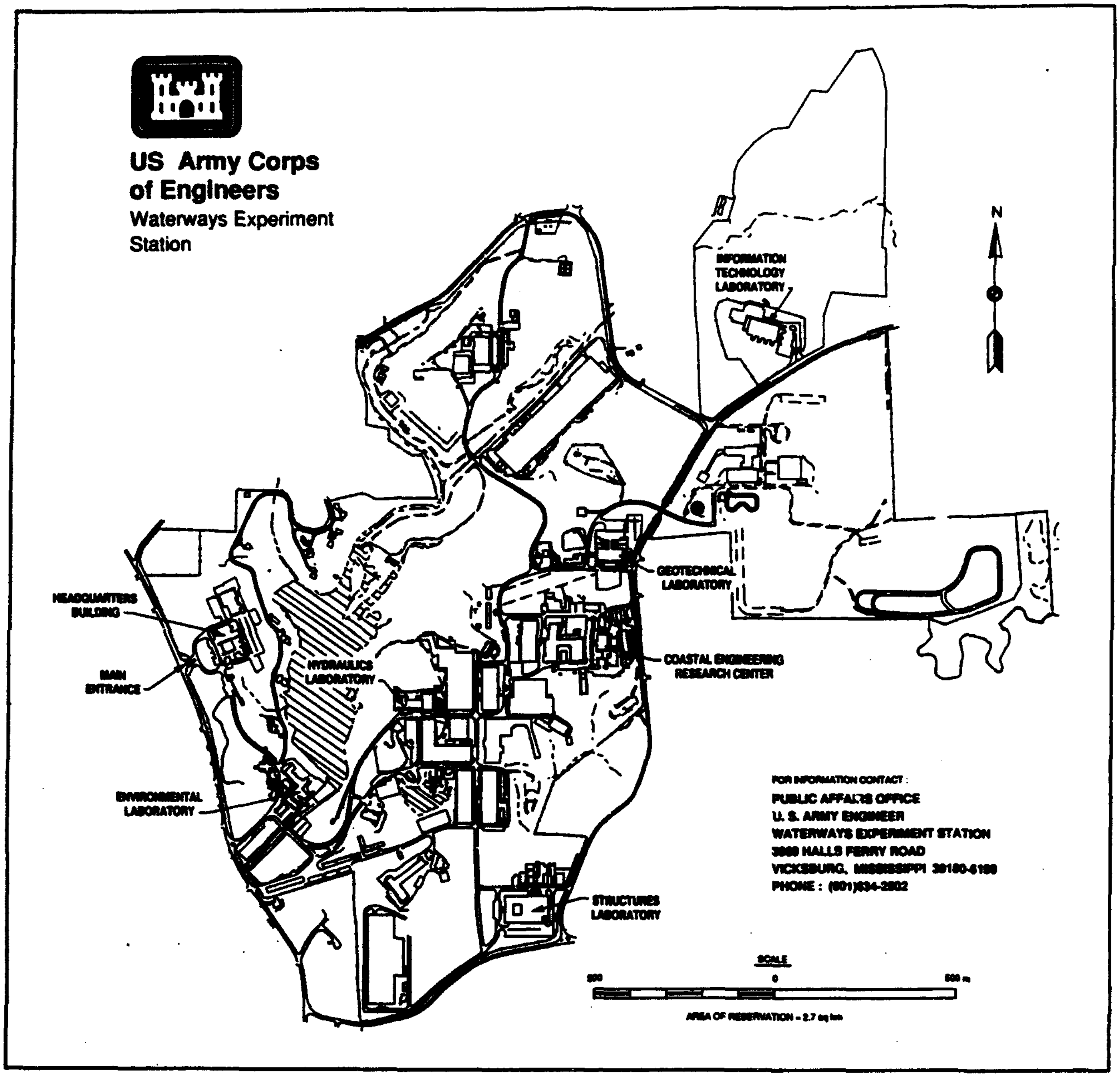
Approved for public release; distribution is unlimited

Prepared for Headquarters, U.S. Army Corps of Engineers
Washington, DC 20314-1000

National Maglev Initiative
Washington, DC 20590

and U.S. Army Engineer Division, Huntsville
Huntsville, AL 35805

Under Project E8691R003, Task 5



Waterways Experiment Station Cataloging-in-Publication Data

Ray, James C.
A model for assessment of dynamic interaction between magnetically levitated vehicles and their supporting guideways / by James C. Ray, Mostafiz R. Chowdhury; prepared for Headquarters, U.S. Army Corps of Engineers, National Maglev Initiative, and U.S. Army Engineer Division, Huntsville.
73 p. : ill. ; 28 cm. -- (Technical report ; SL-94-15)
Includes bibliographic references.
1. Magnetic levitation vehicles. 2. High speed ground transportation -- Design and construction. I. Chowdhury, Mostafiz R. II. United States. Army. Corps of Engineers. III. United States. Army. Corps of Engineers. Huntsville Division. IV. National Maglev Initiative. V. Structures Laboratory (U.S.) VI. U.S. Army Engineer Waterways Experiment Station. VII. Title. VIII. Series: Technical report (U.S. Army Engineer Waterways Experiment Station) ; SL-94-15.
TA7 W34 no.SL-94-15

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Preface

The research reported herein was sponsored by the National Maglev Initiative, Washington, DC, through the U.S. Army Engineer Division, Huntsville, under Project E8691R003, Task 5. Mr. Rick Suever was the Program Monitor.

All work was carried out by Mr. James C. Ray and Dr. Mostafiz R. Chowdhury, Structural Mechanics Divison (SMD), Structures Laboratory (SL), U.S. Army Engineer Waterways Experiment Station (WES), under the general supervision of Mr. Bryant Mather, Director, SL; Mr. J. T. Ballard, Assistant Director; and Dr. Jimmy P. Balsara, Chief, SMD. The work was conducted during the period January-December 1993 under the direct supervision of Mr. Ray.

At the time of publication of this report, Director of WES was Dr. Robert W. Whalin. Commander was COL Bruce K. Howard, EN.

1 Introduction

Background

Practical superconducting magnetic levitation (Maglev) was first invented in the United States in the 1960s. Development of this concept was pursued for a short time in the United States, but in the 1970s, federal funding was cut and thus development in the United States effectively ceased. Foreign governments however continued development and today, both Germany and Japan have working prototypes.

As a result of the evolving foreign technology and increasing transportation needs, the United State's interest in Maglev was renewed in 1990. In December 1990, the National Maglev Initiative (NMI) was formed by the Department of Transportation, the Corps of Engineers, and the Department of Energy. Its purpose was to continue the analyses conducted earlier in evaluating the potential for Maglev to improve intercity transportation in the United States and to determine the appropriate role for the Federal Government in advancing this technology.

As part of its evaluation, the NMI sought industry's perspective on the best ways to implement Maglev technology. They awarded four System Concept Definition (SCD) contracts to teams led by Bechtel Corp., Foster-Miller, Inc., Grumman Aerospace Corp., and Magneplane International, Inc.. These contracts resulted in very thorough descriptions and analyses of four innovative Maglev concepts.

The NMI also formed an independent Government Maglev System Assessment (GMSA) team. This team consisted of scientists and engineers from the U.S. Army Corps of Engineers (USACE), the U.S. Department of Transportation (USDOT), and Argonne National Laboratory (ANL), plus contracted transportation specialists. The team assessed the technical viability of the four SCD concepts, the German TR07 Maglev design, and the French TGV high-speed train (Lever, 1993). Part of these assessments included the use of existing analytical tools to study and compare various high-risk or high-cost concerns related to each system, such as the guideway structures, magnetic suspensions and stray magnetic fields, motor and power systems, and vehicle/guideway interaction. As a result of these assessments, shortcomings were recognized in the state-of-the-art of available analytical tools for this purpose. To address these shortcomings, the NMI funded several

projects to further develop specific analytical tools for use in future evaluations of Maglev designs. The development of the vehicle/guideway interaction (VGI) model reported herein was part of that effort.

VGI refers to the dynamic interaction (coupling) between two separate dynamic systems, the Maglev vehicle and its supporting flexible guideway. The vehicle motion is caused by the roughness and flexibility of its supporting guideway; and the guideway motion is caused by the passage of the vehicle and its time varying suspension forces. In order to accurately predict the performance of either the vehicle or the guideway, the dynamic interaction between the two must be considered.

As a result of previous work related to Maglev, the VNTSC already had a basic computer model for the assessment of vehicular ride quality over flexible and rough guideways. The model was considered "basic" because it only allowed two degrees of freedom for the vehicle: pitch and bounce. It was used successfully in the report by Lever (1993) to perform ride quality assessments.

Also as a result of previous Maglev work (Lever, 1993), the U.S. Army Engineer Waterways Experiment Station (WES) had determined that the ABAQUS Finite Element (FE) program offered the greatest degree of flexibility and accuracy for the dynamic analyses of the diverse range of possible Maglev guideway designs. This program was used successfully in the report by Lever (1993) to perform both two- and three-dimensional static and dynamic guideway analyses.

As seen above, previous work had shown that the analytical capability existed to successfully perform both vehicular ride quality analyses and guideway structural analyses. However, these analyses were completely independent of each other and thus a truly coupled analysis between the vehicle and guideway was not accomplished. The work reported herein describes the development of a model, which intimately links these two analysis procedures together and provides a true VGI analysis.

The model has two distinct applications: It can be used to accurately predict the vehicle ride quality to be expected from a given vehicle and guideway design; and to accurately predict the dynamic deflections and stresses experienced throughout the guideway structure as a result of a vehicle passage. Ride quality results are necessary to design a vehicle suspension system and to determine the guideway stiffness required to meet specific ride quality criteria. Dynamic structural analyses are necessary to produce safe, economical and accurate guideway designs.

Objective

The objective of this work was to develop a methodology (referred to herein as a "model") for the assessment of the dynamic interaction between Maglev vehicles and their supporting guideways. This model will be used to predict the ride quality and stability of Maglev vehicle/guideway designs and the dynamic response of the guideway as a result of vehicle passage.

Scope

The first portion of the project involved the development of a model for the coupling of two separate analytical processes: one for vehicle ride quality analyses and one for dynamic structural analysis. Once the model was developed, efforts were focused on improvement and refinement of the existing analytical processes and verification of the validity and accuracy of the model. The following paragraphs describe these efforts:

The existing vehicle analysis programs only accounted for vehicular pitch and bounce in the vertical plane. These programs were modified to also include roll, yaw, and lateral sway. All work in this area was accomplished by the VNTSC. Therefore, only an overview of this portion of the model is provided herein. A detailed report of this work can be obtained through the VNTSC.

A computer program was written to convert time-varying vehicle bogie forces (determined in the vehicle analysis) into time- and spatially-varying forces on the FE model of the guideway. This program provided the intimate link between the vehicle and guideway analytical processes.

The accuracy and validity of the model were verified with a series of comparative analyses of a simplified system. The system consisted of a single mass (representing the vehicle) suspended by both a primary and secondary suspension (linear springs), which are separated by an intermediate mass (which represents a bogie). This "vehicle" was moved across the flexible beam (representing the guideway) at various speeds. The response of this system was determined and compared using three different solution methods: the VGI model developed herein, a Closed-form analytical solution, and a Finite Element analytical solution. In addition to the verification of the VGI model, the model was used to conduct a series of basic parameter studies on this simplified system. These results were used to study and demonstrate some basic concepts which are important to the design of a Maglev system.

For the purpose of demonstration, the VGI model was applied to the Foster-Miller SCD. This analysis is presented in a step-by-step manner to demonstrate the usage of the VGI Model.

2 VGI Model

Model Description

VGI analysis involves the solution of two completely separate and equally complex dynamic systems: the vehicle with its unique 3-dimensional suspension and control characteristics; and the guideway with its unique 3-dimensional flexibility characteristics. Since the capabilities existed for each of these analyses separately (the vehicle with VNTSC and the guideway with WES), initial consideration was given to the combination of these analyses into a singular all-in-one VGI model. However, this was found to be impractical due to the enormity of each of the calculational processes and the fact that a singular model would severely limit its required flexibility to address all possible Maglev system designs. Therefore, a "VGI Model" was developed which intimately coupled the VNTSC Ride Quality Analysis Code (RQAC) to the WES Guideway Analysis Code (GAC). The function of the VGI Model is depicted in the Figure 1. It is also depicted in flowchart form in Figure 2.

Referring to Figures 1 and 2, the first step in the Model is to build the FE mesh of the guideway (consisting of nodes, elements, and boundary conditions) and perform a modal analysis with the GAC to determine its dynamic mode shapes and frequencies. These guideway properties are used by the RQAC to account for the guideway flexibility in the ride quality analysis (Step 2 in the Model). In addition to vehicular accelerations (used for the ride quality assessment), the RQAC also provides a summary of the time variation in bogie forces at the guideway level as a result of passage over the flexible and randomly-rough guideway. These force-time histories are used to load the FE mesh of the guideway to obtain dynamic deflections and stresses for the guideway (Step 4). Thus, the complete circle is made and the dynamic interaction between the vehicle and guideway is accounted for in both analyses. The loop is "closed" (i.e. checked for accuracy) in Step 5. The guideway deflections predicted by the guideway equation in the RQAC are compared to those obtained from the GAC. If they do not compare to within reasonable limits (+/- 15 percent), the process must be repeated using more guideway mode shapes in the RQAC. The following sections describe in greater detail the components of the VGI Model and the methods used to link them together.

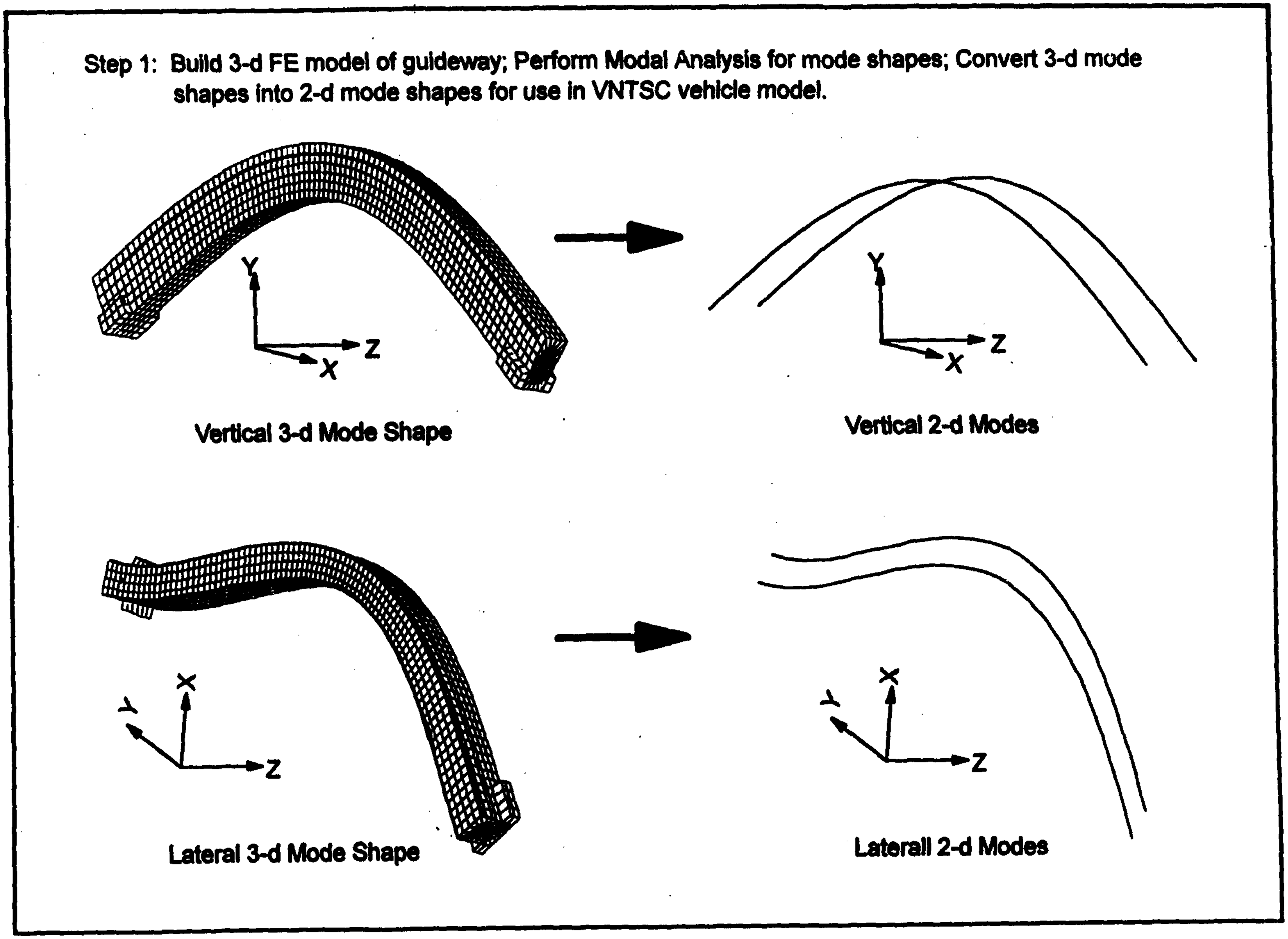


Figure 1. Pictorial of VGI Model (Continued)

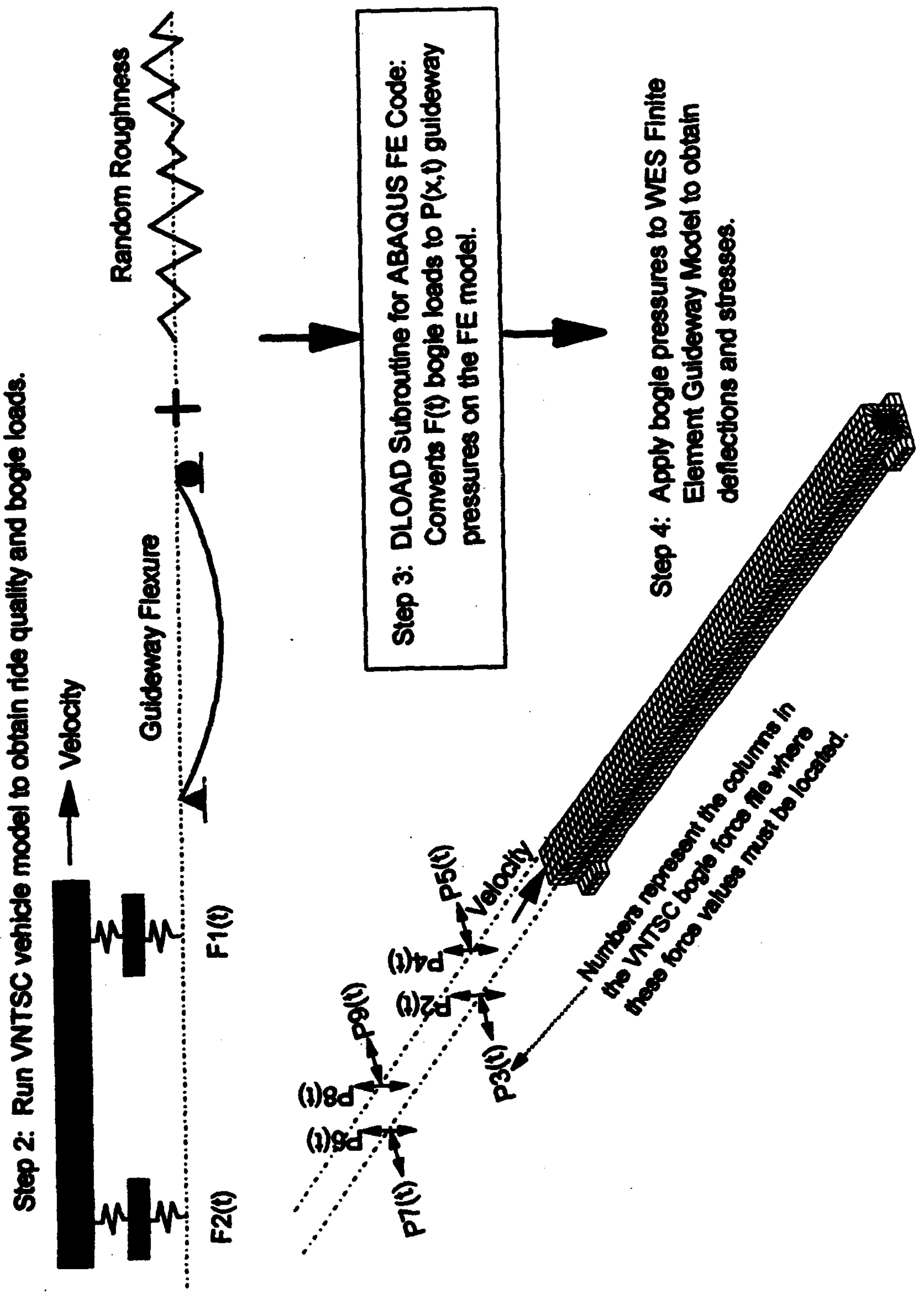


Figure 1. (Concluded)

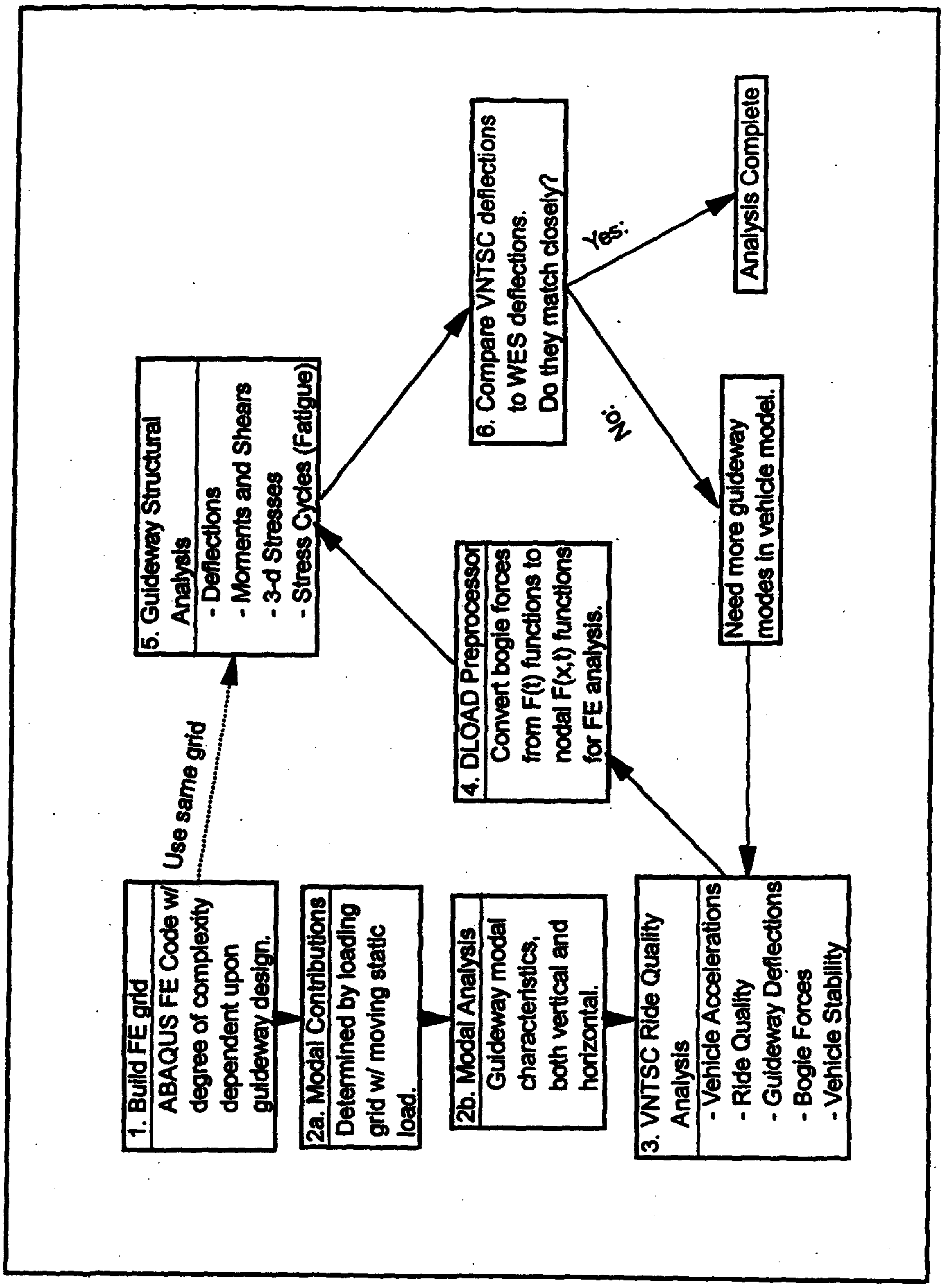


Figure 2. Flowchart of VGI Model

Vehicle Ride Quality Analysis Code (RQAC)

All work on the RQAC was accomplished by the VNTSC. Therefore, only an overview of this portion of the model is provided in the following paragraphs. A detailed report of the work accomplished on the RQAC can be obtained through the VNTSC.

Figure 3 shows the general vehicle-suspension-guideway interaction problem. As the vehicle moves along the guideway, it is acted upon by external forces and by the suspension system forces which cause linear and rotational accelerations of the vehicle. If the vehicle has appreciable flexibility, the forces may also cause significant deformation of the vehicle body. The suspensions react to the vehicle body and guideway surface motions, where the latter include geometric irregularities, intentional camber, and elastic deformations, and produce suspension forces acting on the vehicle and on the guideway. The guideway beams deflect dynamically in response to the moving, unsteady suspension forces and the reaction forces and moments generated at the support. Finally, the support motions are determined by the support and foundation dynamic characteristics and the guideway beam forces and moments. This strongly coupled situation is extremely complicated in general and simplifying assumptions must be made in order to make engineering analysis feasible.

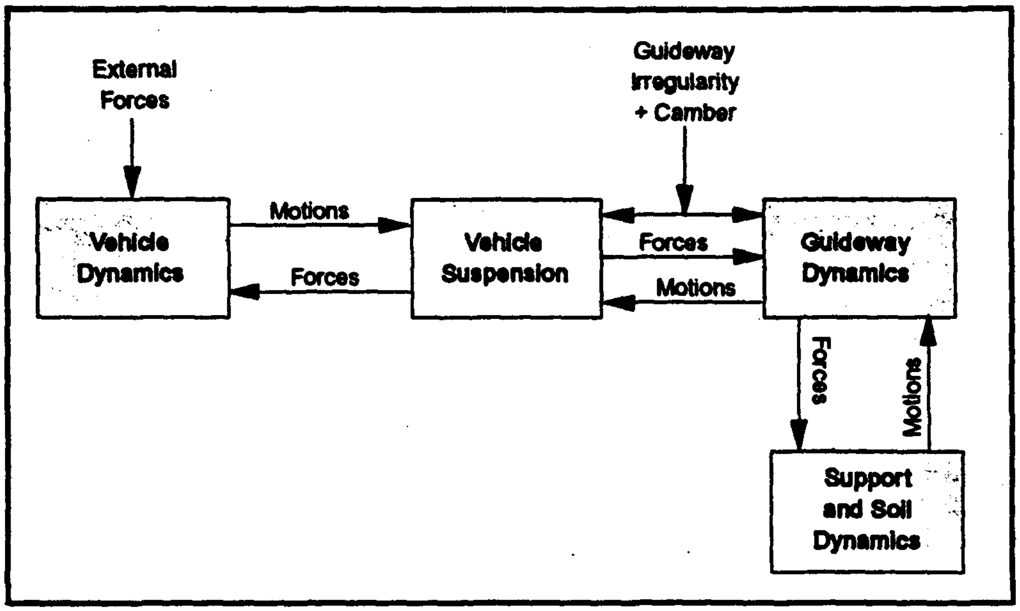


Figure 3. General vehicle-guideway interaction solution

Figure 4 shows a simplified model which has been employed widely in the literature for vehicle-guideway dynamic analysis (Richardson and Wormely 1974). For this model, only motions in the vertical plane are considered and the vehicle is assumed to have a rigid body with mass m_v and moment of inertia J about its center of mass. The front and rear suspensions each include a primary suspension (composed of a spring, k_p, and damper, c_p) which acts directly against the guideway, and a secondary suspension consisting of a spring, k_s, and damper, c_s, or including active elements. A bogie with mass, m_b, separates the primary and secondary suspensions.

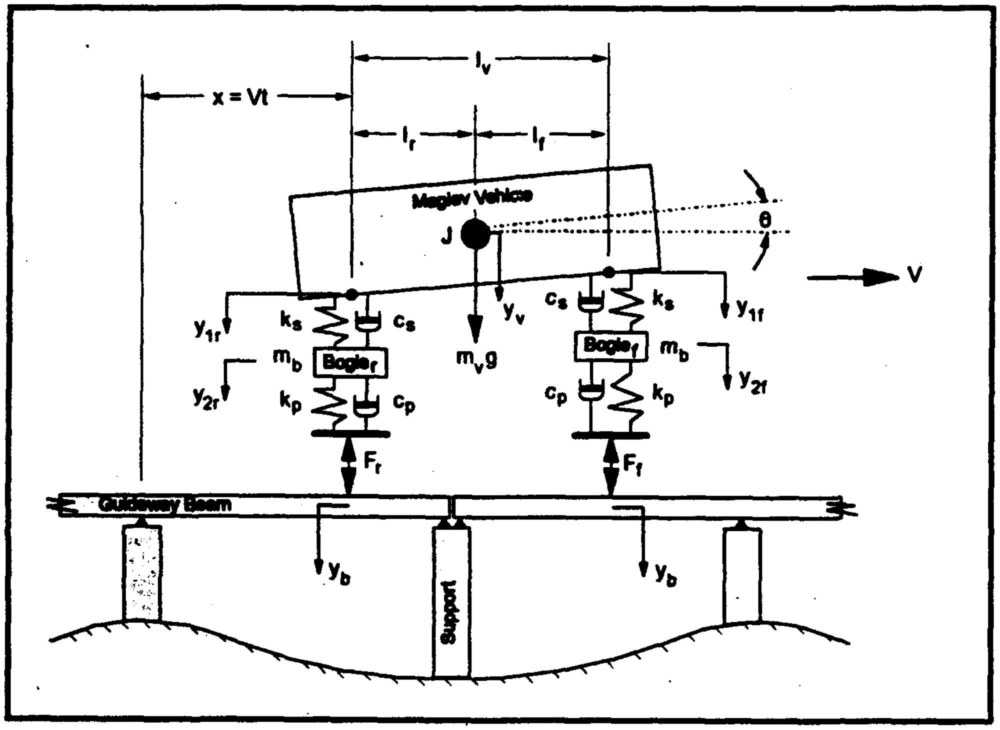


Figure 4. Simplified Maglev system

The governing equations for the vehicle are:

$$F_f = \frac{m_v g}{2}\left[\frac{m_b}{m_v} + \frac{2l_r}{l_f + l_r} - \frac{m_b}{m_v}\frac{\ddot{y}_{2f}}{g} + \frac{2l_r}{l_f + l_r}\left(-\frac{\ddot{y}_v}{g} + \frac{J\ddot{\theta}}{m_v g l_r} + \frac{F_e}{m_v g} - \frac{T_e}{m_v g l_r}\right)\right] \tag{1}$$

$$F_r = \frac{m_v g}{2}\left[\frac{m_b}{m_v} + \frac{2l_f}{l_f + l_r} - \frac{m_b}{m_v}\frac{\ddot{y}_{2r}}{g} + \frac{2l_f}{l_f + l_r}\left(-\frac{\ddot{y}_v}{g} + \frac{J\ddot{\theta}}{m_v g l_f} + \frac{F_e}{m_v g} - \frac{T_e}{m_v g l_f}\right)\right] \tag{2}$$

The external force, F_e, and torque, T_e, may include the coupling forces between multiple vehicles.

The suspensions can be described by appropriate dynamic relations between the force exerted on the guideway, F, the motion of the suspension attachment point on the vehicle, y_1, and the apparent motion of the guideway surface below the suspension. This later consists of the guideway elastic deflection, y_b, and the geometric irregularity or misalignment, yo.

$$F = f\left(y_1, y_b + y_0\right) \tag{3}$$

Note that this force does not include the static weight, $m_b g/2$, which must be added to the dynamic force, F, to obtain the total suspension forces F_r or F_f for use in Equations (1) and (2).

Both active and passive suspensions can be modeled. That for an active suspension will be described in the VNTSC Report. For the simpler case of a passive suspension having a primary spring of stiffness, k_p, a secondary stiffness, k_s, and secondary damping (in parallel), c_s, the relationship shown above becomes

$$\overline{F} = k_p \frac{\left[(c_s s + k_s)\overline{y}_1 - \left(\frac{m_b}{2} s^2 + c_s s + k_s\right)\left(\overline{y_b} + \overline{y}_0\right)\right]}{\left(\frac{m_b}{2} s^2 + c_s s + k_s + k_p\right)} \tag{4}$$

where $m_b/2$ is the individual suspension unsprung mass, s is the Laplace transform operator, and $\overline{()}$ indicates the Laplace transform.

The geometrical irregularity and misalignment of the guideway is described by the guideway profile $y_0(x)$ when n vehicle loads are present and will consist in general of at least four components: beam distortion due to dead weight or thermal strain, misalignment of spans, random surface roughness, and camber. The first two effects will contain components strongly periodic at span wavelength while the third will normally be broad band with irregularity amplitudes increasing with wavelength. Camber refers to an intentional initial profile given to a beam in an attempt to compensate for beam sag or for vehicle-induced deflections. The methods for incorporating all of these effects into the analytical process are discussed in greater detail in the following section.

The beam (refer to Figure 4) is described by the Bernoulli-Euler partial differential equation

$$EI\frac{\partial^4 y_b}{\partial x^4} + ky_b + b\frac{\partial y_b}{\partial t} + \rho a\frac{\partial^2 y_b}{\partial t^2} = p(x,t) \tag{5}$$

where E is the elastic modulus, I the moment of inertia, ρ the mass density and a the cross-sectional area of the beam. The term ky represents a distributed

elastic restraint (elastic foundation) and is zero for discretely supported beams. The damping coefficient b represents a dissipative force acting between the beam and ground and is usually used to approximate all damping effects in the structure. The suspension force is described by a spatial distribution of pressure which moves along the beam with the vehicle velocity, V.

The modal analysis method (Biggs 1964) is used in the Vehicle Model to solve the guideway dynamics problem. The Bernoulli-Euler equation is used as the basis for the modal solution technique and the space and time varying motion $y_b(x,t)$ of the guideway is represented as a summation of the natural mode solutions of the equation

$$y_b(x,t) = \sum_{m=1}^{\infty} A_m(t)\phi_m(x) \tag{6}$$

where A_m are the modal amplitudes, independent of x, and Φ_m are the modal shape functions which are orthogonal over the interval $0 < x < L$ and functions only of x.

The mode shapes $\Phi_m(x)$ are determined from the natural unforced vibration of the span and are affected by the support boundary conditions at the ends of the beams and at intermediate supports in the case of multiple-span beams. The time-varying amplitude functions A_m depend on the forcing function p(x,t) in Equation (5) while the natural modal shape functions satisfy the homogeneous part of the equation. By substituting (6) into (5) with p(x,t) = 0, a set of modal shape functions Φ_m and corresponding undamped natural frequencies ω_m can be determined for any prescribed support conditions and tabulated. Using the orthogonality property of Φ_m and assuming the shape functions are normalized so that

$$\frac{1}{L}\int_0^L \phi_m^2(x)dx = 1 \tag{7}$$

a set of ordinary differential equations is derived for each modal coefficient of the form

$$\ddot{A}_m + 2\xi_m\omega_m\dot{A}_m + \omega_m^2\dot{A}_m = \frac{1}{\rho a L}\int_0^L p(x,t)\phi_m(x)dx \tag{8}$$

where ξ_m is the modal damping ratio and $(\dot{\ })$ indicates d/dt. Since the guideway loading p(x,t) usually consists of a specified pressure profile which moves with the vehicle velocity, v, and varies in magnitude with time, the right side of (8) can be integrated explicitly to obtain a function of time only. Equations (8), (1), (2), and (3) are then solved simultaneously to find the guideway and vehicle motions and associated forces. For the closely coupled case herein where p(x,t) depends strongly on vehicle and suspension motion, a point-by-point integration of the equations is required where the timesteps must be small enough to capture all of the important frequencies within the system.

Note that additional bogies and vehicles will greatly increase the number of simultaneous equations and thus the complexity of the problem solution.

Equation (8) can only be used to describe the dynamic response of very simple structures, where the modal frequencies, ω_m, and modal shape functions, $\phi_m(x)$, can be easily defined with conventional closed-form solutions. The modal parameters for complex guideway structures, such as those defined by the SCD contractors, cannot be defined in this manner. Therefore, the Finite Element Method (FEM) must be used to define these mode shapes in discrete terms, which can replace the $\phi_m(x)$ term in Equation (8). The FEM can also provide accurate values for modal frequencies, ω_m.

Whereas Equation (8) describes the dynamic equation of motion for the guideway using the method of Modal Superposition, the following is the FEM equation of motion for the guideway:

$$[M]\{\ddot{y}\}+[C]\{\dot{y}\}+[K]\{y\}=\{f(t)\} \tag{9}$$

where [C] is constrained to be Rayleigh damping:

$$[C]=\alpha[M]+\beta[K] \tag{10}$$

Both sides of the equation can be multiplied by the transpose of a matrix, $[H]$ of the mode shapes (eigenvectors). This matrix is not required to be square, and should only contain the mode shapes associated with the lower mode shapes, i.e. the mode shapes that contribute the most to the displacement of the guideway. This matrix should contain the mode shape associated with the lowest frequency (eigenvalue) up to the mode shape associated with a frequency about three times greater than the highest crossing frequency to be analyzed. This process is shown in Equation (11):

$$[H]^t[M]\{\ddot{y}\}+[H]^t[C]\{\dot{y}\}+[H]^t[K]\{y\}=[H]^t\{f(t)\} \tag{11}$$

The coordinate transformation described in Equation (12) below can be substituted in equation (11), resulting in Equation (13).

$$\{y\}=[H]\{z\} \tag{12}$$

$$[H]^t[M][H]\{\ddot{z}\}+[H]^t[C][H]\{\dot{z}\}+[H]^t[K][H]\{z\}=[H]^t\{f(t)\} \tag{13}$$

$[H]^t[M][H]$ is the modal mass, while $[H]^t[C][H]$ is the modal damping and $[H]^t[K][H]$ is the modal stiffness. These modal parameters are outputs of the FE analysis, along with the mode shapes. Equation (13) can be rewritten as

$$[M^*]\{\ddot{z}\}+[C^*]\{\dot{z}\}+[K^*]\{z\}=[H]^t\{f(t)\} \tag{14}$$

In general, the matrices of Equation (14) will be substantially smaller than the matrices of equation (9). Equation (9) may have matrices that are several thousand rows with an equal number of columns, while Equation (14) may have matrices that are only tens of rows with an equal number of columns. Equation (14) is relatively simple to integrate as the modal matrices $[M^*]$,$[C^*]$, and $[M^*]$ are all diagonal. Equations (14), (1), (2), and (3) are then solved simultaneously using a point to point integration scheme to find the guideway and vehicle responses and associated forces. Using simpler terms to represent the vehicular equations of motion, the simultaneous equations for the VGI system can be represented as follows:

$$\begin{bmatrix} M_v & 0 \\ 0 & M^* \end{bmatrix} \begin{Bmatrix} \ddot{Z}_v \\ \ddot{Z}_g \end{Bmatrix} + \begin{bmatrix} C_v & 0 \\ 0 & C^* \end{bmatrix} \begin{Bmatrix} \dot{Z}_v \\ \dot{Z}_g \end{Bmatrix} + \begin{bmatrix} K_v & 0 \\ 0 & K^* \end{bmatrix} \begin{Bmatrix} y_v \\ y_g \end{Bmatrix} = \begin{Bmatrix} F_v \\ F_g \end{Bmatrix} \quad (15)$$

where

$$F_g = [H]^t \{f(t)\} \quad (16)$$

The nodal forces {f(t)} are a vector of the forces generated by the vehicle acting on the nodes of the guideway FE model. The loads generated by the vehicle move as the vehicle traverses the guideway. In general, the location of these loads at specific integration time increments will not coincide with nodes of the FE model. The loads generated by the vehicle model range from pressure forces of arbitrary distribution to point loads. The methodology used to distribute the loads to surrounding nodes of the FE model is described in the following portions of this chapter.

Guideway Analysis Code (GAC)

General

The guideway dynamics of the VGI Model are determined using the general-purpose ABAQUS Finite Element Code (HKS Inc 1992). The finite element method of structural analysis is well documented in the literature and thus its theory will not be discussed in detail herein. Gallagher (1975) provides an excellent overview of the FE methodology. The basic concept of the method, when applied to problems of structural analysis, is that a continuum (the total structure) can be modeled analytically by its subdivision into regions (the finite elements). The behavior of each of these regions is described by a separate set of assumed functions representing the stresses or displacements in that region. These sets of functions are often chosen in a form that ensures continuity of the described behavior throughout the complete continuum. Much like the alternative procedures for the accomplishment of numerical solutions for problems in structural mechanics, the finite element

method requires the formation and solution of systems of algebraic equations. The special advantages of the method reside in its suitability for automation of the equation formation process and in the ability to represent highly irregular and complex structures and loading situations.

As previously described, the GAC is used for two specific purposes within the VGI Model. It is first used to determine the discrete modal parameters which will be used in the modal superposition portion of the RQAC to account for guideway flexibility. Then, once the ride quality analysis is accomplished and dynamic vehicle forces obtained, these forces are used by the GAC to calculate three-dimensional deflections and stresses in the guideway due to the vehicle passage. The following paragraphs describe the components of the GAC and its adaptation to the VGI model.

FE Mesh Creation

In order to perform an FE analysis of any structure, it must first be descritized into a "mesh" of finite elements. The boundaries of the elements are defined by nodes. The elements composing the mesh of a given structure may be of any size and number. However, the refinement of the mesh (i.e. number of elements and nodes composing the mesh) must be carefully chosen in each case in order to obtain accurate results with the greatest economy of computational requirements. Numerous guidelines are provided in the literature for the effective idealization of structures with finite elements. Meyer (1987) provides an excellent discussion of this topic for conventional structures. For use in the GAC, the size of the elements must be based on several other primary factors:

The FE mesh must be of such refinement that its solution will correctly reproduce those characteristic mode shapes of the real structure which are likely to be excited by the loads. For the case of a Maglev vehicle traversing an accurately-constructed (i.e. smooth) guideway, the loading frequency is mainly a function of the vehicle crossing speed and bogie spacing. It will depend to a limited extent upon the bogie load variation resulting from such things as guideway roughness and misalignment. However, these load variations will be of very high frequency, generally small in magnitude for smooth guideways, and thus will have little affect on the overall guideway response. Richardson and Wormely (1974) indicate that for guideways with k equal spans, the number of modes important for accurate displacement calculations will be equal to k and that for bending moment and stress can be greater than $3k$. Thus, for a single-span guideway, the mesh should be sufficiently refined to accurately represent the first 3 bending modes (all in the same global direction) if moments and stresses are to be determined. A minimal mesh of this type is depicted in Figure 5.

If a study of localized deflections and stresses is desired, a more

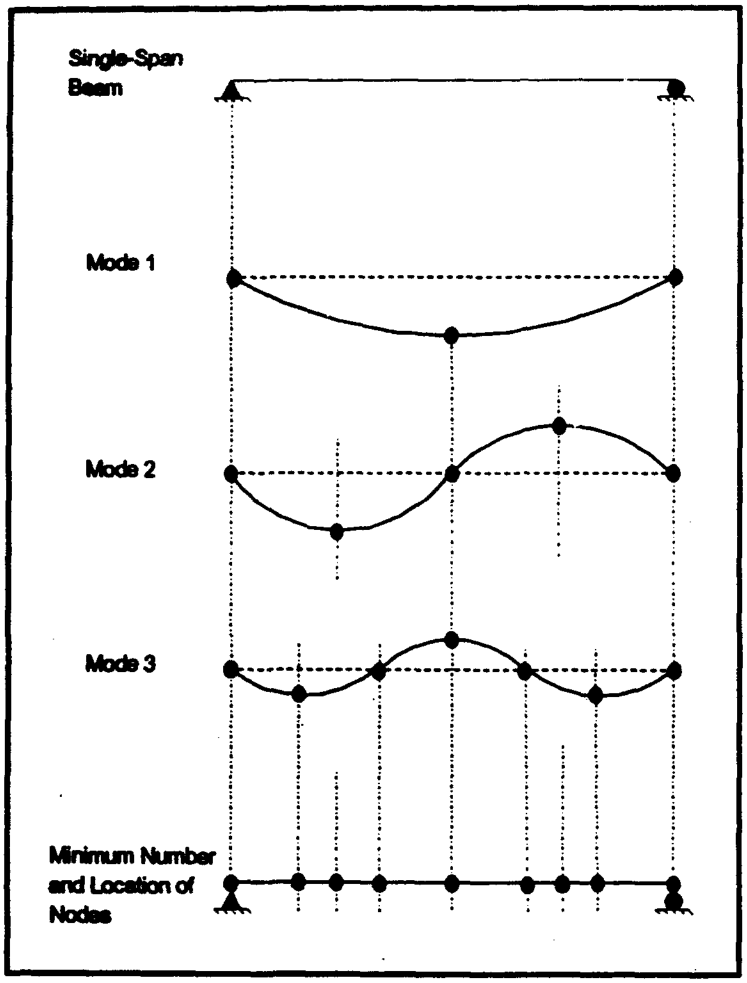


Figure 5. Minimal mesh to represent a simple beam

refined mesh must be used. Note however that the high frequency modes associated with localized responses will only be excited if the passing vehicle has high frequency loads associated with it.
Significant loads of this nature are often not produced from soft-sprung Maglev-type suspensions. Therefore, complex localized deflections and stresses can often be most efficiently studied using a very refined mesh in combination with static load applications. The dynamics of the system can be studied with a considerably coarser mesh.

Based on the above discussion, the mesh density for use in the GAC can generally be relatively coarse, resulting in minimal computer computational requirements. However, an additional criterion unique to the GAC is that the length of any loaded finite element must always be less than 2 times the shortest bogie length on the loading vehicle. This is a requirement set forth by the discrete loading methodology of the DLOAD subroutine and will be discussed in detail in that later section.

Modal Analysis

Once the FE mesh of the guideway is constructed. the ABAQUS FE code is used to obtain all of the necessary 3-d mode shapes and modal parameters. These data are then used in the modal superposition portion of the RQAC to obtain the total solution to the dynamic equation of motion (Equation (9) above). *In the modal superposition method*, the total response is obtained by summing the forced responses of each of the individual natural modes of free vibration of the guideway. Modal analysis is discussed in numerous references. However, the following paragraph provides a brief explanation of the process:

The problem of free vibration requires that the force vector {F} be equal to zero in the guideway equation of motion (See Equation (9) above). The equation thus becomes:

$$[M]\{\ddot{y}\}+[C]\{\dot{y}\}+[K]\{y\}=\{0\} \tag{16}$$

For free vibrations of the undamped structure, the solutions of Equation (16) will have the form

$$y_i = a_i \sin(\omega t-\alpha) \quad i = 1,2,\ldots\ldots,n$$

or in vector notation

$$\{y\}=\{a\}\sin(\omega t-\alpha), \tag{17}$$

where a_i is the amplitude of motion of the ith coordinate and n is the number of degrees of freedom. The substitution of Equation (17) into Equaiton (16) gives:

$$-\omega^2[M]\{a\}\sin(\omega t-\alpha)+[K]\{a\}\sin(\omega t-\alpha)=\{0\},$$

or rearranging terms:

$$[[K]-\omega^2[M]]\{a\}=\{0\}, \tag{18}$$

which for the general case is a set of n homogeneous (right-hand side equal to zero) algebraic system of linear equations with n unknown displacements a_i and an unknown parameter ω^2. The nontrivial solution, that is, the solution for which not all $a_i = 0$, requires that the determinant of the matrix factor of {a} be equal to zero. In this case,

$$|[K]-\omega^2[M]|=0. \tag{19}$$

In general, Equation (19) results in a polynomial equation of degree n in ω^2

which should be satisfied for n values of ω^2. This polynomial is know as the "characteristic equation of the system". For each of these values of ω^2 satisfying the characteristic Equation (19), Equation (18) can be solved for a_n, in terms of an arbitrary constant.

A modal analysis using the ABAQUS FE code will result in all of the individual 3-d mode shapes (an infinite number actually exist) which make up the total dynamic response to loading. However, the RQAC only utilizes a minimal number of guideway modes in order to simplify the integration of the VGI equations. Therefore, the mode shapes chosen for use in the RQAC must be those which have the greatest effect on the vehicle response. This can be accomplished through use of modal participation factors. These factors are part of the ABAQUS modal output and reflect the degree to which each mode participates in the total flexural response in each of the six principal directions (x, y, z, θ_x, θ_y, θ_z). Once the important mode shapes are chosen, they must be provided in a usable form for the RQAC. As seen in Figure 1, the FE guideway modes are 3-dimensional (3-d). Yet the RQAC requires 2-dimensional (2-d) guideway modes. Since the vehicle will only be in contact with the guideway along specific slide lines (i.e. where the levitation magnets slide along the guideway), the displacements along these lines are all that is actually required by the RQAC. Since the magnet lines on each side of the guideway are generally parallel, this will effectively allow the reduction of a 3-d mode shape to a 2-d shape at each of the magnet slide lines as shown in Figure 1. The displacement coordinates of the 2-d shapes are provided only at the nodes of the FE mesh since these are the only locations where loads are allowed to act upon the FE mesh. By providing the predominant 2-d modes in all directions (i.e. vertical, horizontal, and torsional) to the RQAC, the appropriate 3-d response of the vehicle will be excited, and as a result, the appropriate 3-d loadings will be re-applied to the guideway FE model for the guideway analysis.

Structural Analysis

The second function of the GAC is to perform a dynamic analysis of the guideway structure to determine its response to the vehicular loadings. The results from this analysis are dynamic deflections and stresses at any location on the guideway structure. The general principals of Finite Element analysis of structures are well documented in the literature and thus will not be discussed herein. The ABAQUS code was used herein for all FE analyses.

Although the FE analysis of these types of structures is relatively straight-forward, the loading of the FE mesh for the analysis is far from straight-forward. The mesh must be loaded with the resulting time-dependent bogie forces from the RQAC. This is difficult since the ABAQUS code does not allow the application of a moving, time-varying load. The RQAC provides separate force-time histories for each bogie of the vehicle and for each principal direction of loading. The force-time history represents the variation in bogie force as it traverses the guideway during the ride quality analysis.

Therefore, these forces must be applied to the guideway mesh at not only the correct time, but also at the appropriate location on the mesh. In effect, a set of fixed-location node or element force-time histories must be produced that represent the moving and time-varying bogie loads. This was accomplished by developing a special subroutine, named "DLOAD", that is executed along with the ABAQUS FE code. It acts basically as a co-processor to calculate the element loadings at each time increment within the FE analysis.

The use of a DLOAD subroutine is an option normally available in the ABAQUS code to allow the application of non-uniform pressure loadings to elements. For a dynamic FE analysis, the DLOAD routine is called at the beginning of each time increment for each integration point within each loaded element. A specific pressure loading is calculated and applied over the area of the element enc mpassed by that integration point. Since each element has multiple integration points, this results in a non-uniform pressure distribution over the element. Thus, a time-varying non-uniform pressure can be applied to any desired element within the FE mesh. The DLOAD subroutine and several supporting computer codes, as written specifically for use in the GAC of the VGI Model, are provided in hardcopy form in Appendix A. The following paragraphs describe the logic of the DLOAD routine.

Once the FE mesh of the guideway is constructed, selected elements within the mesh are defined as "loaded" elements; i.e. the bogie loads will be effectively swept across and applied to these elements at the desired vehicular velocity. These elements are defined with the ELSET option within the ABAQUS input file. Figure 6 represents a row of loaded elements within an FE mesh. Although only one row is shown, any number of rows may be loaded. The elements may be of any type (beam, solid, shell, etc.). All FE calculations are performed at integration points within the elements. The 8-noded elements in Figure 6 each have 9 unique integration points on their surfaces, arranged according to a Gaussian distribution. Once calculated, all DLOAD pressures will be applied to the portion of the element area encompassed by that specific integration point, which for the case in Figure 6, will be 1/9 of the total element face area.

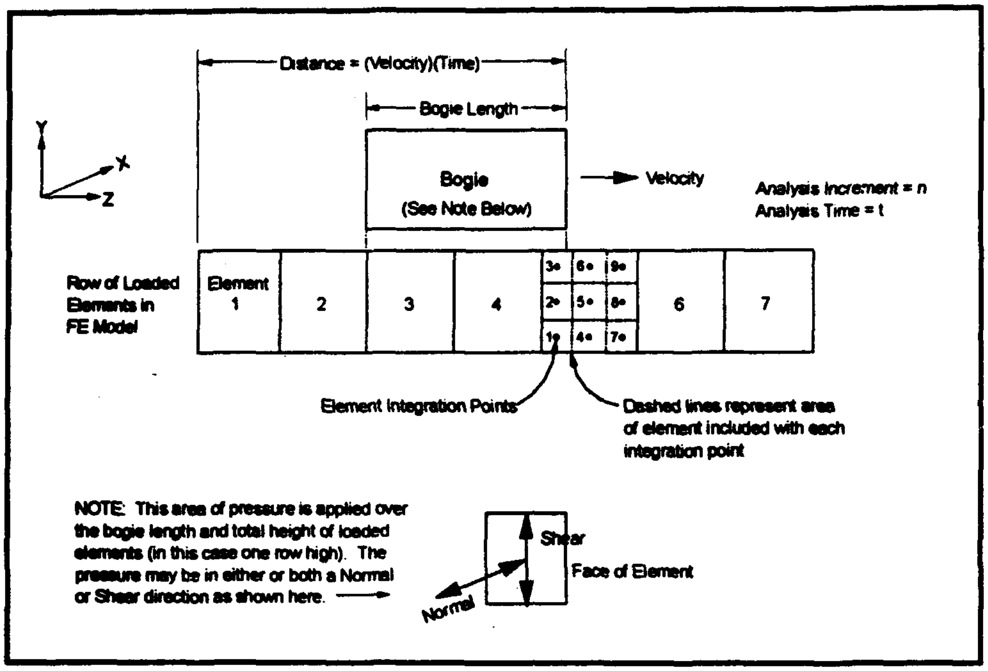


Figure 6. Demonstration of DLOAD Subroutine

The "DLINGEN" program (short for DLOAD Input Generator) was written to automate the input of specific vehicle information to the DLOAD routine. Its source code is also provided in Appendix A. The input variables are discussed as follows: As shown in Figure 7, the vehicle is defined as a succession of bogie sets, where a set consists of one bogie on each side of the vehicle at the same z-location along the length of the vehicle (i.e. right and left bogies). Set number 1 starts at the beginning of the guideway mesh (i.e. at z=0) and the successive sets are behind it at specifically-defined z-locations (dependent upon the vehicle design). Each of the bogie sets have finite lengths and heights over which the bogie pressure is applied to the mesh. Since pressures are applied to the loaded element sets, the bogie height (or width, depending upon orientation) must correspond to the total height (or width) of the loaded element set. Pressures can be applied in either or both the vertical (y-axis) or transverse (x-axis) direction(s). Depending upon the orientation of the loaded elements, this will translate into normal or shear loadings on the elements. The variable "NORMAL" must be defined for the DLOAD routine as shown in Figure 8 to distinguish the direction of loadings on the FE mesh.

During a dynamic analysis, the ABAQUS code calls the DLOAD subroutine at the beginning of each time increment for each integration point of the loaded elements. With time, t, and vehicular velocity as constants, the DLOAD routine calculates the exact location of each of the bogie sets and determines which integration points should be loaded during that timestep. For example, referring to Figure 6, all of the integration points of Elements 3 and 4 and the first 3 integration points of Element 5 are covered by Bogie 1 at time, t. The bogie pressure (bogie force at time, t, divided by bogie area) will

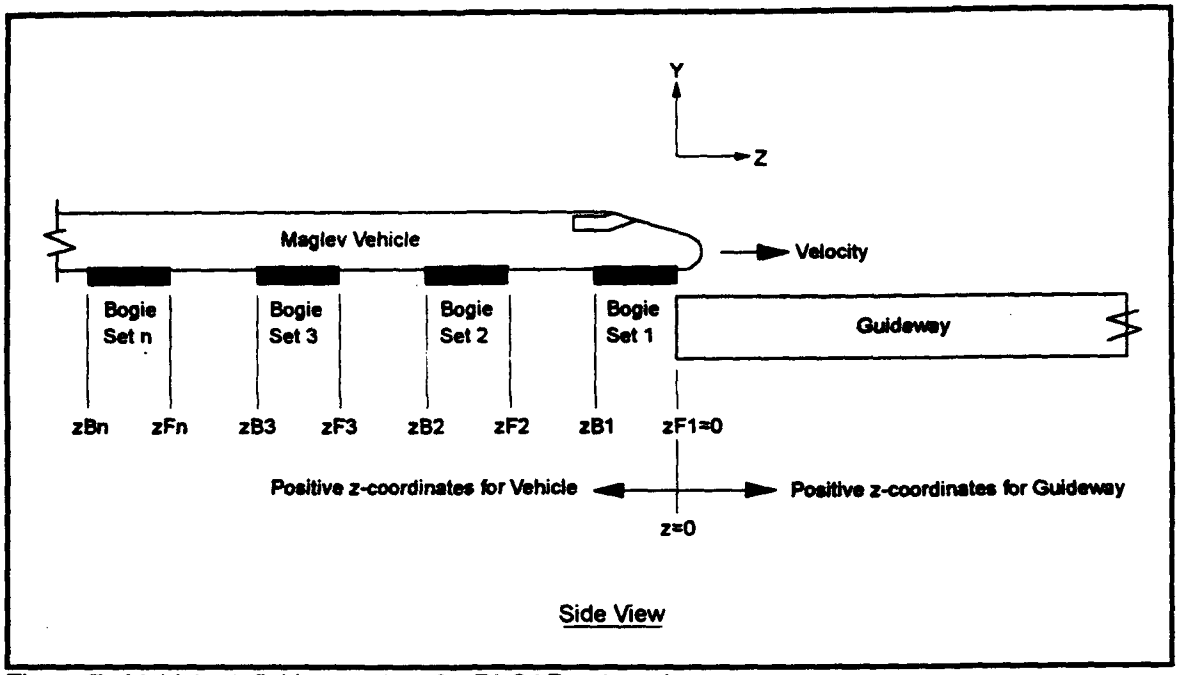


Figure 7. Vehicle definition method for DLOAD subroutine

thus be applied to these points only. At the next time increment, Bogie 1 will be further along, covering more points within Element 5 and less points in Element 3. The bogie force (from which bogie pressure is calculated) at the specific time, t, is obtained from the force data file (normally supplied from the RQAC), as shown in Appendix A. Each row of this file must have the format of: time, shear $force_n$, normal $force_n$, up to n bogie sets. The bogie force components at each time should be given in columnar format in the order shown in Figure 8. The force values in this file are given at specific times and thus a linear interpolation must be made to get the force at a specific time, t. Normal and shear loads are applied simultaneously to the same point by making two DLOAD calls from the ABAQUS input file, one for JLTYP=20 (normal loading) and one for JLTYP=2 (shear loading).

An integration point is considered to have pressure on it if its z-coordinate is less than or equal to that for the front of a bogie set and is greater than or equal to that for the rear of the bogie set. In Figure 6, all of the integration points of elements 3 and 4, and the first three of element 5 are covered and thus loaded. No rise or decay time is given to the loadings. This causes a slight delay in the loading of the integration points and thus a slight lag in the structural response. For example (referring to Figure 6), the integration points of Element 1 will not see any loading until the first bogie has traveled a distance of the element length divided by 6. If the element lengths are relatively short, this effect should be negligible.

From Figure 6, it can be seen that the analytical timesteps (Δt) must be small enough to match the frequency of the integration points within the elements. Specifically, Δt should always be less than or equal to the

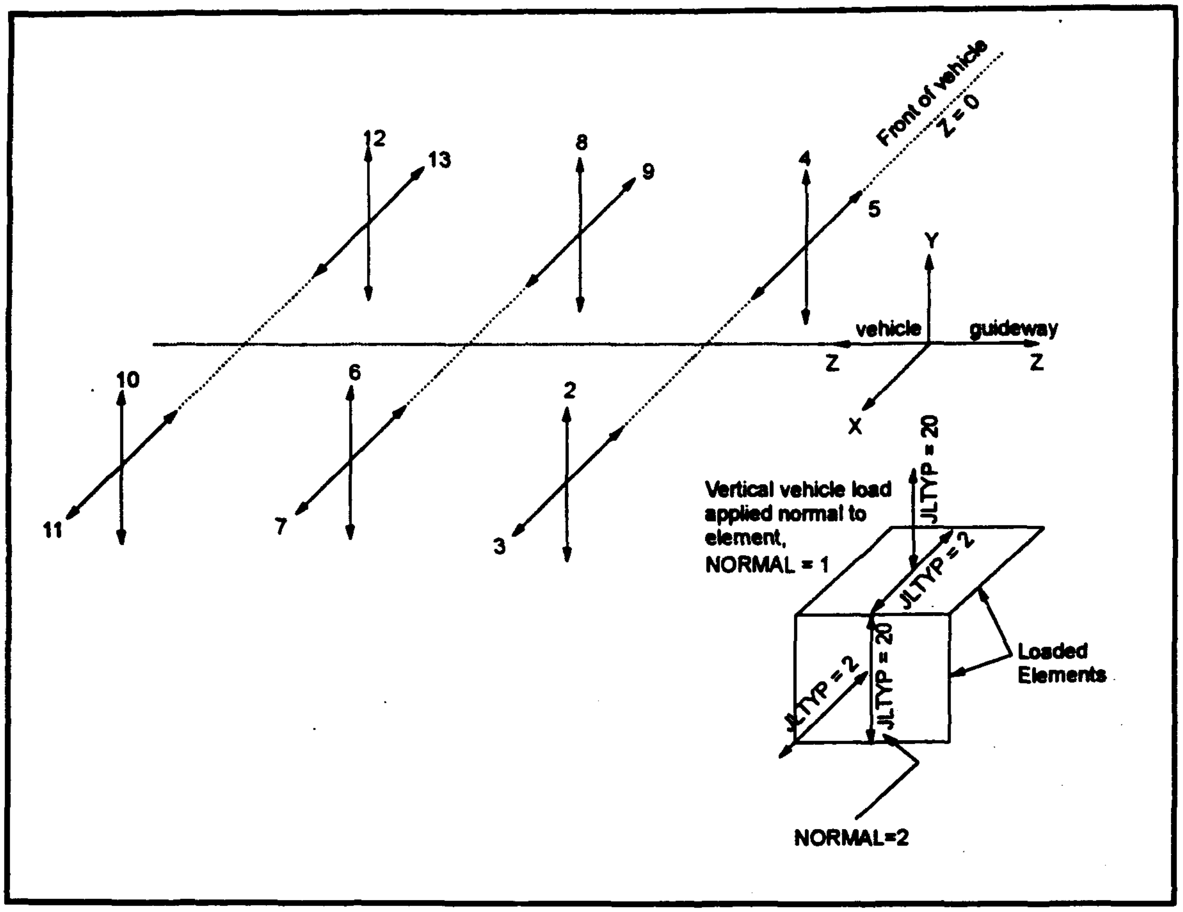


Figure 8. Ordering of force columns and definition of the variable "NORMAL" for the DLOAD Subroutine

integration point spacing divided by vehicle velocity. In addition, the logic of the DLOAD routine requires that the bogie length should never be less than approximately 1/2 element length in order to insure that a bogie does not end up in between two sets of integration points during a timestep, thus producing no load on the element. For normal Maglev-type vehicles, these "rules of thumb" should not present a problem.

Solution Verification

Step 5 in the VGI Model (refer to Figure 2) is to check the accuracy of the VGI analysis. This is done by comparing guideway deflections determined from both the RQAC (Step 2) and the GAC (Step 4). If these deflections check to within reasonable limits (+/- 15%), it should be apparent that all of the important guideway bending modes were included in the RQAC and its guideway deflections, and thus its ride quality and vehicular force predictions, were accurate. If deflections do not closely compare, the VGI analysis should

be redone, beginning with Step 2 and including additional guideway modes in the RQAC.

It is important to note that even though guideway deflections are calculated by the RQAC, it cannot be used to determine the stresses in the guideway. The stresses are related to the spatial derivative of the displacement. Numerically evaluating derivatives amplifies errors, and the small errors allowed when only a few modes are used to calculate displacement (as done in the RQAC) can cause substantial errors if derivatives are taken. As a result, the follow-on dynamic FE guideway analysis is required for stress analysis.

Limitations of VGI Model

As with any numerical model, assumptions and simplifications were necessary in the development of the VGI model. All of these items were described in the previous sections. Their impact on the accuracy and validity of the model is addressed in the following paragraphs:

Coupled/Uncoupled Natural Frequencies

Although not discussed in detail in this report, several assumptions are made in the RQAC which should be presented herein for a thorough understanding of the VGI Model. The following discussion comes mostly from Tyrell (1993).

The forcing functions on the right-hand side of the vehicle and guideway equations (Equation 15 herein) can be expanded into terms that include the vehicle suspension stiffness and the guideway mode shape. Consequently, they are dependent upon the vehicle and guideway position and velocity. If expanded, they could be combined with similar ones on the left-hand sides of the equations. However, it is substantially less difficult computationally and in the formulation of the specific system equations to keep these terms on the right-hand sides, thus keeping the natural frequencies independent of each other . This simplification should be valid for most Maglev systems since the vehicle suspension is soft in relation to the stiffness of the guideway.

The model shown in Figure 9 was used by Tyrell (1993) to check the validity of this simplification. Neglecting damping, expanding forcing terms, and moving the appropriate factors to the right-hand side results in the following equation:

$$\begin{Bmatrix} \ddot{y}_v \\ \ddot{y}_g \end{Bmatrix} + \begin{bmatrix} \omega_v^2 & -\omega_v^2\phi \\ -R\omega_v^2\phi & \omega_g^2 + R\omega_v^2\phi^2 \end{bmatrix} \begin{Bmatrix} y_v \\ y_g \end{Bmatrix} = \begin{Bmatrix} 0 \\ -Rg\phi(x) \end{Bmatrix} \tag{20}$$

where:

$\omega_v = \sqrt{\dfrac{k_v}{m_v}}$ (Uncoupled vehicle natural frequency)

$\omega_g = \sqrt{\dfrac{EI}{\rho A}}\left(\dfrac{\pi}{L}\right)^2$ (Uncoupled guideway first-mode natural frequency)

$R = \dfrac{m_v}{\rho AL}$ (Vehicle/Guideway mass ratio)

$\phi = \dfrac{\sqrt{2}}{2}\sin\dfrac{\pi x}{L}$ (Guideway first-mode spatial shape)

The natural frequencies of the system are:

$$\Omega_g^2 = \omega_g^2\left[\frac{1}{2}+\left(\frac{1}{2}\phi^2R+\frac{1}{2}\right)\left(\frac{\omega_v}{\omega_g}\right)^2 - \frac{1}{2}\sqrt{1+\left(2\phi^2R-2\right)\left(\frac{\omega_v}{\omega_g}\right)^2+\left(\phi^4R^2+\phi^2R+1\right)\left(\frac{\omega_v}{\omega_g}\right)^4}\right]$$

(21a)

$$\Omega_v^2 = \omega_v^2\left[\frac{1}{2}\left(\frac{\omega_g}{\omega_v}\right)^2+\frac{1}{2}\phi^2R+\frac{1}{2}+\frac{1}{2}\sqrt{\left(\frac{\omega_v}{\omega_g}\right)^4+\left(2\phi^2R-2\right)\left(\frac{\omega_g}{\omega_v}\right)^2+\left(\phi^4R^2+\phi^2R+1\right)}\right]$$

(21b)

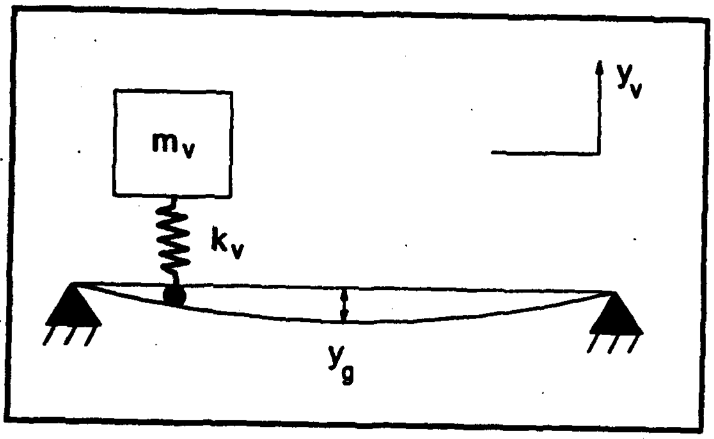


Figure 9. Simplified vehicle/guideway model

The coupling terms in Equation (21) all include the spatial mode shape ϕ and the vehicle/guideway mass ratio, R. The value of ϕ is zero when the vehicle is located at the ends of the beam and is maximum when the vehicle is located in the center of the beam. Consequently, the coupling of the vehicle and guideway is weakest when the vehicle is near the end of the beam and strongest when near the center of the beam. In addition, the coupling between the vehicle and guideway is weak when R is small (i.e. with a relatively light vehicle compared to the guideway) and when the uncoupled natural frequencies are well separated.

Figure 10 shows the shift in natural frequencies for the system when the vehicle is at the center of the beam (that is, ϕ is maximum) for three different values of R. The figure shows that the uncoupled natural frequencies at the center of the beam are reasonably accurate for values of R less than 0.75 when the uncoupled guideway natural frequency and the uncoupled vehicle natural frequency differ by 50 percent or more. It also shows that for values of R less than 0.25, the uncoupled natural frequencies at the center of the beam are reasonably accurate when the uncoupled guideway natural frequency and the uncoupled vehicle natural frequency differ by 10 percent or more.

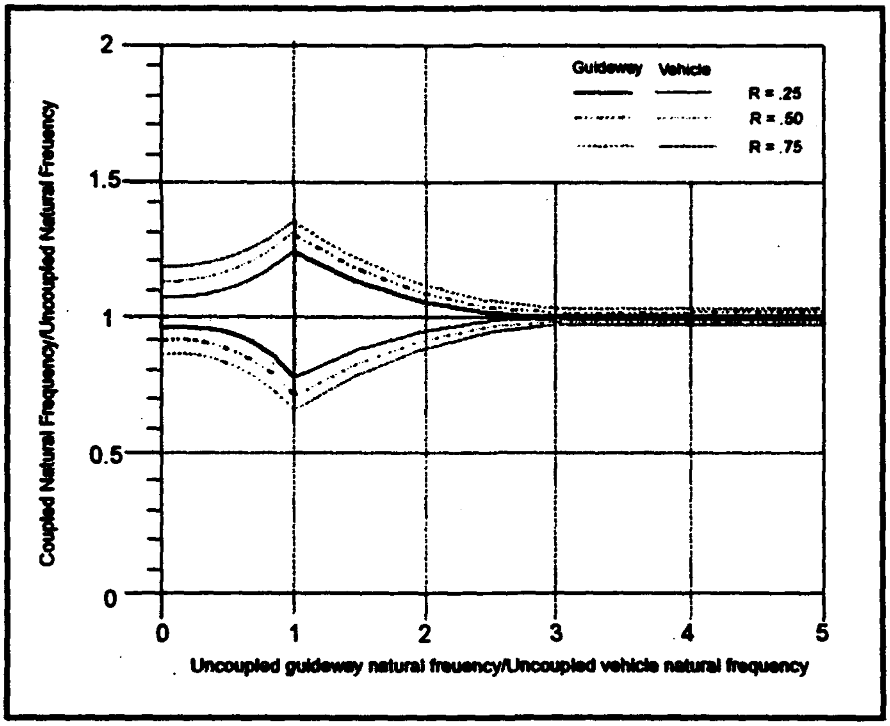


Figure 10. Shift in natural frequencies with relative guideway stiffness (Tyrell 1993)

Figure 11 shows the influence of the vehicle position on the coupled natural frequencies associated with the guideway and the vehicle when the weight of the vehicle is half the weight of the beam, and the uncoupled natural frequency of the beam is 1.25 times the uncoupled natural frequency of the vehicle. In this case, the coupling of the vehicle and the guideway is strongest when the vehicle is in the center-third of the beam.

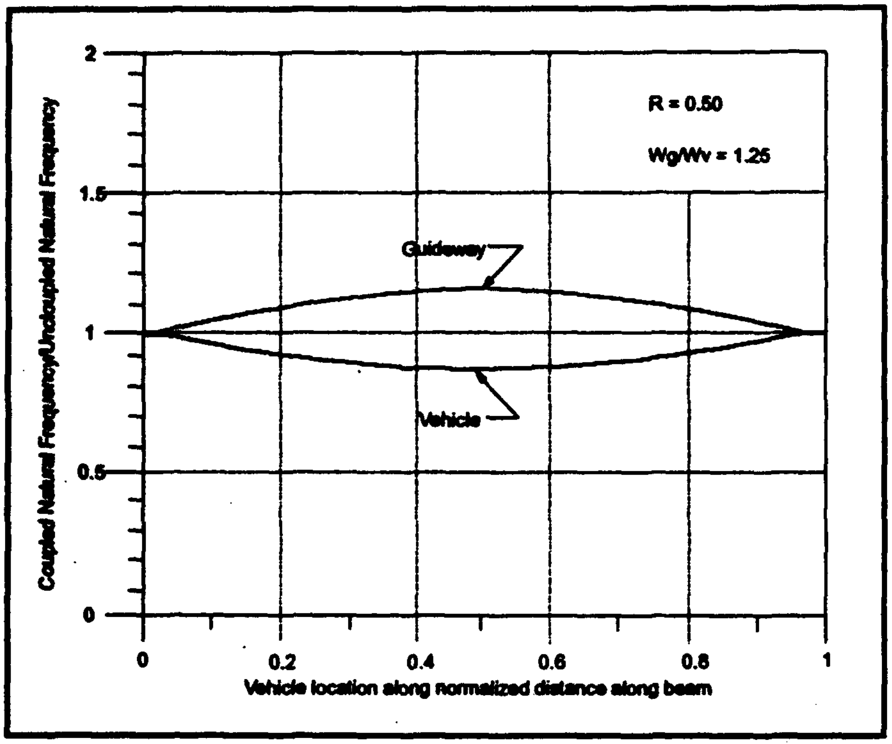


Figure 11. Influence of vehicle position on coupled frequencies (Tyrell 1993)

In general, those vehicle configurations which incorporate a primary and secondary suspension may be expected to have less than 25 percent of the vehicle mass acting on a guideway section. Because of ride quality requirements, vehicle configurations which incorporate a single suspension may be expected to have vehicle natural frequencies which are substantially lower than the guideway natural frequencies. Table 1 lists the parameters for several specific Maglev systems. For these systems, the maximum error in the eigenvalues is less than 10 percent. For the TR07 system, the primary suspension natural frequency is the same as the guideway first-mode natural frequency. However, the primary suspension/guideway mass ratio is 0.092. Because of this small ratio, the dynamic coupling of the vehicle and guideway is very weak. For the Foster-Miller SCD, the guideway is massive compared to the vehicle, and consequently the dynamic coupling between the guideway and vehicle is weak. The dynamic coupling is minimized in the Grumman SCD because the guideway first-mode natural frequency is approximately half the natural frequency of the vehicle suspension and the vehicle/guideway mass ratio is moderately low.

Table 1
Maglev System Parameters (Tyrell 1993)

System	Primary Suspension Nat. Freq. (Hz)	Secondary Suspension Nat. Freq. (Hz)	Guideway First Mode Nat. Freq. (Hz)	Vehicle/Gdwy Mass Ratio
TR07	6.0	1.0	6.0	0.65
Foster-Miller	4.3	1.2	6.4	0.08
Grumman	9.1	N/A	4.4	0.28

Therefore, the solution approach used in the RQAC appears to be valid for the evaluation of ride comfort and the determination of the low frequency (less than about 15Hz) variations in the vertical forces acting between the vehicle and guideway. As shown above, the dynamic coupling between the vehicle and guideway is reduced by the relationship between their natural frequencies, the ratio of their masses, and the spatial mode shapes of the guideway. For vehicle/guideway combinations that do not fall within the criteria discussed above, a different numerical solution technique may be required.

Guideway Modal Contributions

As previously shown in Equation (11), only a minimum number of guideway bending modes should be utilized in the RQAC in order to simplify the integration of the VGI equations. These modes must be carefully chosen in order to insure that all load-excitable modes are included. Tyrell (1993) states that the mode shape matrix should include the mode with the lowest bending frequency and all other modes with frequencies up to about three times greater than the highest vehicle crossing frequency to considered. A similar guideline is also recommended by Richardson and Wormely (1974). As an example of this guideline: The Foster-Miller SCD (Foster-Miller 1993) has a vehicular bogie spacing of approximately 27m and a guideway span length of 27m. For a vehicular velocity of 140m/s, the crossing frequency will be approximately 5 Hz. Thus, all guideway modes with frequencies less than 15 Hz should be included in the RQAC solution.

For complex guideways such as that for the Grumman SCD (Grumman 1993), the above guidelines for choice of mode shapes may not be sufficient. Another method is the use of modal participation factors, as described previously in the "Modal Analysis" section of this report. These factors are standard output from the ABAQUS FE modal analysis. They directly reflect the degree to which each mode contributes to deflection in each direction.

Multispan Analyses

The RQAC can be used to predict ride quality over numerous consecutive guideway spans. This is a very important capability since vehicular accelerations can in some cases be multiplied or damped with each successive span passage. This variation in vehicular accelerations will also affect the bogie forces that are transmitted to the guideway. Thus, in some cases, multiple spans should also be considered in the GAC to insure that the maximum vehicular loadings are applied to the guideway. However, the GAC is not as conducive to multispan analyses since each span must be discretized into finite elements, each of which will increase computer CPU requirements. Several different options can be used to address this limitation if multiple spans must be considered in the GAC:

If a relatively coarse FE mesh can be used (refer to the "Mesh Creation" section above for this criteria), multiple spans can likely be analyzed without over-taxing the computer. If a refined FE mesh is required, only a limited number of span lengths will be possible for the GAC. In this case, the bogie force input file (output from the RQAC) can be scanned to determine the times at which the maximum forces occurred and only this portion applied to the limited number of spans in the GAC.

3 VGI Model Verification

Approach

The accuracy and validity of the VGI model was verified with a series of comparative analyses of the simplified system shown in Figure 12. The system consisted of a single mass (representing the vehicle) suspended by both a primary and secondary suspension (linear springs), which are separated by an intermediate mass (which represents a bogie). This "vehicle" was moved across the flexible beam (representing the guideway) at various speeds. The system was simplified in that it only considered motion in the vertical plane and only represented one support point (i.e. one bogie set) on a Maglev vehicle, which in reality would be supported in multiple locations.

The response of this system was determined and compared using three different solution methods: the VGI model, a closed-form analytical solution, and a finite element analytical solution. In addition to the verification of the VGI model, the model was used to conduct a series of basic parameter studies on this simplified system. These results were used to study and demonstrate some basic concepts which are important to the design of a Maglev system.

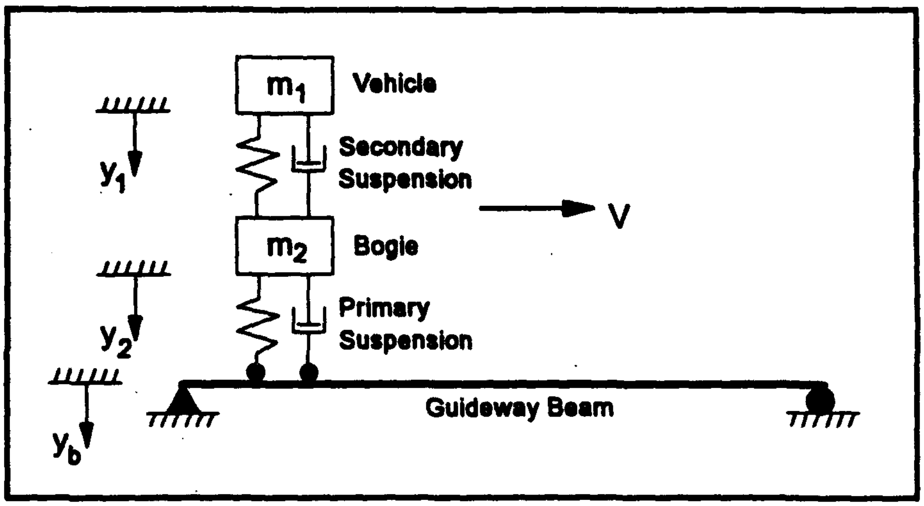


Figure 12. Simplified maglev system

Closed-form Solution

The system in Figure 11 can be represented by the free-body diagrams shown in Figure 13. Note that all displacements are measured from the static equilibrium position.

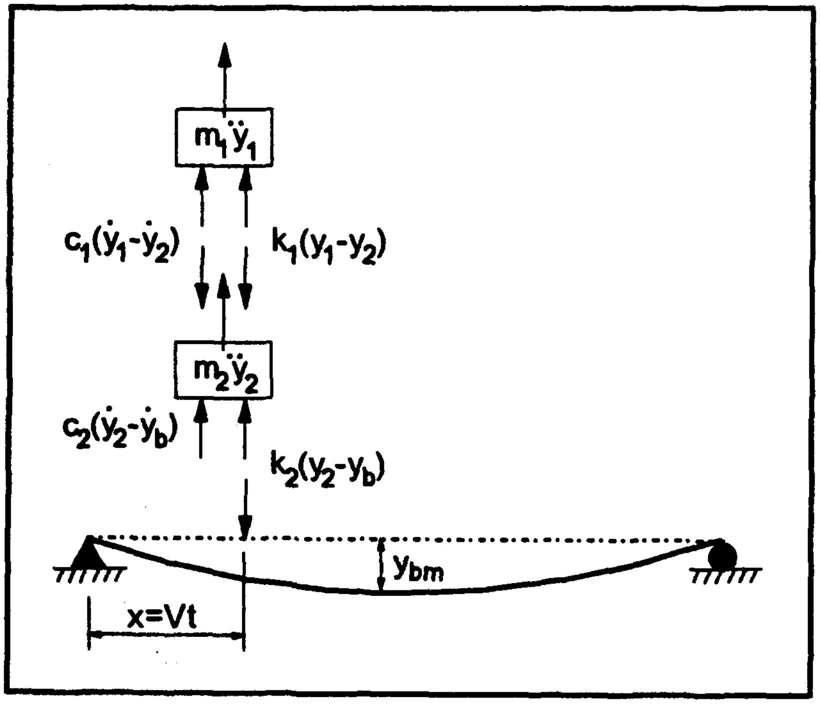


Figure 13. Free-body diagram for the system

The dynamic deflection, y_b, of the beam can be computed by using the method of "Modal Superposition". The Bernoulli-Euler equation is used as the basis for this technique and the space and time varying motion $y_b(x,t)$ of the beam is represented as a summation of the natural mode solutions of the equation

$$y_b(x,t) = \sum_{r=1}^{N} A_r(t)\phi_r(x) \tag{22}$$

where A_r are the modal amplitudes, independent of x, and ϕ_r are the modal shape functions which are orthogonal over the length of the beam and functions only of x. For the simply-supported beam in Figure 12, the modal shape, $\phi_r(x)$ is

$$\phi_r(x) = \sin\left(\frac{r\pi x}{L}\right) \tag{23}$$

Using Modal Superposition and neglecting damping, the uncoupled equation of motion for a simply-supported beam subjected to a moving, constant-velocity point load, q, is,

$$\ddot{A}_r(t) + \omega_r^2 A_r(t) = \frac{q\phi_r(x)}{\int_0^L m\phi_r^2(x)dx} \tag{24}$$

where ω_r and ϕ_r are the modal frequency and mode shape, respectively, for the nt. mode; x is the instantaneous position of the point load measured from the end of the span; m is the mass per unit length. For a simple beam, the integral in Equation (24) becomes ml/2. The point force, q, applied to the beam may be expressed as:

$$q_{dynamic} = k_2(y_2 - y_b)$$

$$q_{total} = k_2\left[y_2 - \sum_{r=1}^{N} A_r \sin\left(\frac{r\pi vt}{L}\right)\right] + m_1 g + m_2 g \tag{25}$$

Substituting Equation (25) into Equation (24), the uncoupled Equations of Motion in terms of the modal displacement are obtained. The Equations of Motion for the system in Figure 13 are given in Matrix form as follows:

$$[m]\{\ddot{y}\} + [k]\{y\} = \{f(t)\} \tag{26}$$

Where:

$$[m]\{\ddot{y}\} = \begin{bmatrix} m_1 & 0 & 0 \\ 0 & m_2 & 0 \\ 0 & 0 & \frac{mL}{2} \end{bmatrix} \begin{Bmatrix} \ddot{y}_1 \\ \ddot{y}_2 \\ \ddot{y}_{bm} \end{Bmatrix}$$

$$[k]\{y\} = \begin{bmatrix} k_1 & -k_1 & 0 \\ -k_1 & (k_1 + k_2) & -k_2 \sin B \\ 0 & -k_2 \sin B & k_2 \sin^2 B + \frac{mL}{2}\omega_1^2 \end{bmatrix} \begin{Bmatrix} y_1 \\ y_2 \\ y_{bm} \end{Bmatrix}$$

$$\{f(t)\} = \begin{Bmatrix} 0 \\ 0 \\ (m_1 + m_2) g \sin B \end{Bmatrix}$$

$$B = \frac{(\pi vt)}{L}$$

Note that for calculation simplicity, only the 1st modal contribution to the beam response is included. For simply-supported beams, this is a reasonable simplification.

The equations in the matrix were solved using the Runga-Kutta method (Scarborough 1966). The variables were defined to somewhat represent a typical Maglev-type system as follows:

Vehicle: $m_1 = 22{,}630$ kg;
Secondary Suspension: $k_1 = 0.60\text{x}10^6$ N/m, $c_1 = 0$;
Bogie Mass: $m_2 = 7{,}380$ kg;
Primary Suspension: $k_2 = 2.65\text{x}10^6$ N/m, $c_1 = 0$;
Beam: L= 25m, $\rho = 2{,}400$ kg/m^3, E= 30x10^9 N/m^2
Width=1.0m, Height= 2.96m

Finite Element Analysis

The ABAQUS FE code was used to analyze the system shown in Figure 12 with all of the same variable definitions as described above. The guideway was modeled in two different ways: as a 2-D simple beam constructed from 2-node linear beam elements; and as a 3-D slab with a vertical flexural stiffness equivalent to the 2-D beam model, constructed from 8-node doubly-curved shell elements. The equivalent mass and moment of inertia of the slab were obtained by equating the modal frequencies of both systems. This equivalency insured similar dynamic behavior for both systems as demonstrated later.

The vehicle and bogie were modeled as lumped masses coupled together with linear springs. Two sets of vehicle and bogie elements (each with half the mass of those in the 2-D model, and connected to each other with "Link Elements") were used in the 3-D model, with one set at each edge of the slab, in order to load the slab more uniformly. The elements were moved over the guideway models at constant velocities. Their interaction with the guideway was modeled with 2- and 3-D Slide Line Contact elements, an option in ABAQUS which provides a frictionless, moving interaction surface between two components. Both FE models are depicted in Figure 14. The ABAQUS input file for this analysis is provided in Appendix A.

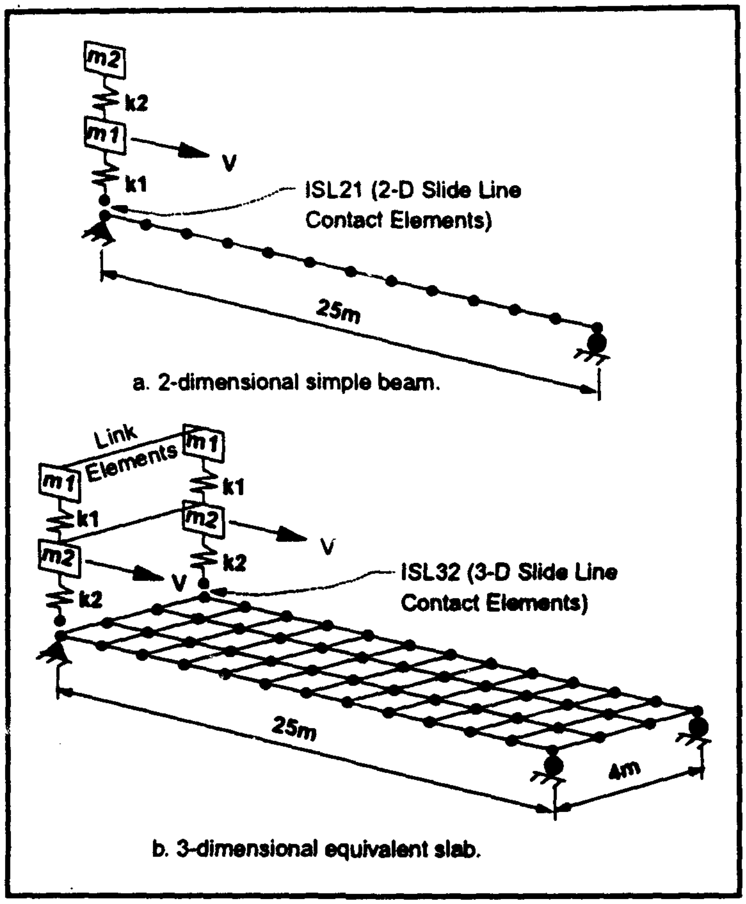


Figure 14. Finite element models

VGI Model

The response of the slab shown in Figure 14 was also obtained using the VGI Model as described in Chapter 2. Since a vehicle ride quality analysis was not desired from this analysis, Steps 2 and 3 of the Model (refer to Figure 2) were not performed. The time-varying vehicular bogie forces at the guideway level, as required for input to the DLOAD routine (Step 4 of the Model), were obtained from the previously discussed Closed-form solution of this problem. These forces will be shown and discussed in the "Results" section of this Chapter. For a normal VGI analysis using this model, the bogie forces will be obtained from the VNTSC Ride Quality Model. The FE model of the slab was the same as that used in the above-described FE analysis.

As previously described, the DLOAD routine serves to convert time-varying bogie forces into time- and spatially varying pressures on specific elements of the FE model of the structure. For the slab in Figure 14, the two outermost rows of elements (along the length) were defined as the loaded elements. The bogie loads were spread out over a length (defined as "bogie

length" in DLOAD) of 1m and a width of 1m, the width being determined by the widths of the loaded elements. It is important to point out that the loads were applied as point loads in the other two analytical methods described above, whereas these were applied as pressure loadings over a 1m by 1m area on each side of the slab. This made a slight difference in the slab responses and will be pointed out in the "Results" section of this chapter.

Results

Midspan beam deflections predicted by the four comparable analytical methods (one Closed-Form, two FE using slide lines, and one with the VGI Model) are compared in Figure 15. A vehicle velocity of 83m/s was used in each case. As expected, the responses for both the beam and slab FE models were almost identical, with the slight variation likely due to additional 3-D modes in the slab model . Likewise, the Closed-Form Solution method showed a slightly lower maximum response because only one beam flexural mode was considered in that analysis.

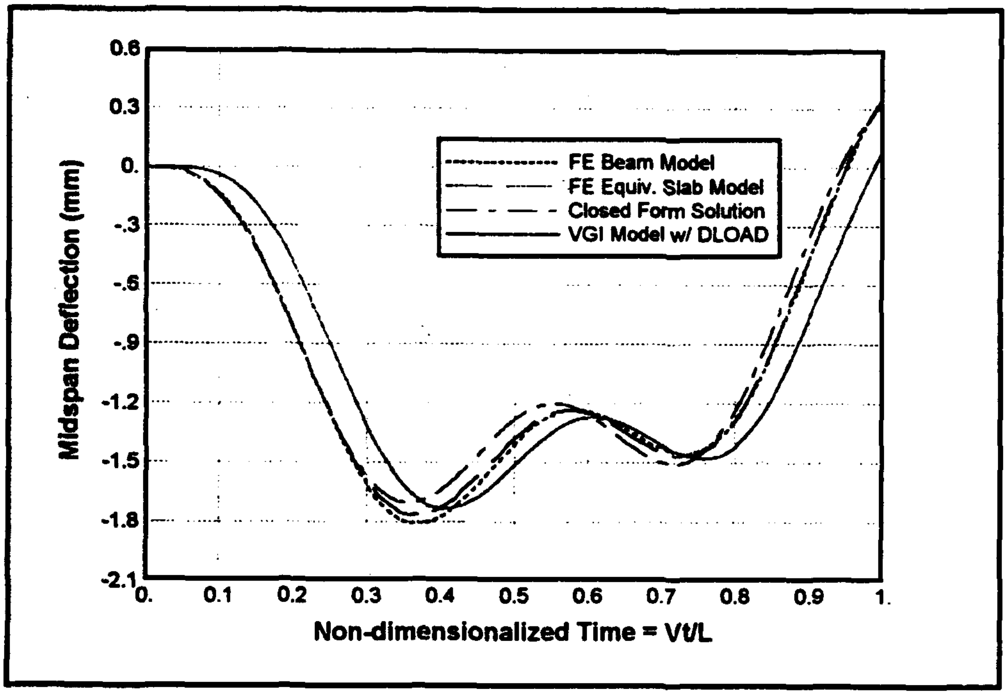


Figure 15. Comparison of analytical methods

The main purpose of conducting both the FE analysis and the Closed Form analysis was to verify the accuracy of the VGI Model described in this report. As can be seen in Figure 15, the results from the VGI Model compared very closely to all of the other analytical results. The slab response predicted by the VGI Model lagged approximately 0.008 seconds behind that

from the other analyses. With a vehicle velocity of 83m/s, this corresponds to a distance of 0.66m, which is about half an element length in the FE model of the slab. This lagging response is due to the manner in which the integration points within the finite elements are loaded by the DLOAD routine and the fact that no rise or decay time is used for the loadings (Refer to the description of the DLOAD routine in Chapter 2). Although this causes a slight lag in the response of the structure, it only amounts to a very small error and thus should produce no real problem in an actual VGI analysis. However, if necessary, additional work can be done on the DLOAD routine to add the effect of load rise and decay on the loaded areas.

Figure 16 shows the 2-D Closed-Form Solution response of the entire system (i.e. beam, bogie, and vehicle) for a vehicle passage velocity of 140 meters per second (mps). The differences in the deflection magnitudes and lagging response times of the bogie and vehicle show the combined effect of their associated masses and suspension stiffnesses. The importance of conducting a VGI analysis over a series of multiple spans is demonstrated in this plot by the fact that both the vehicle and bogie masses are significantly displaced from their original positions as they reach the end of the first span. This would require a completely different set of initial conditions for the analysis of the next span.

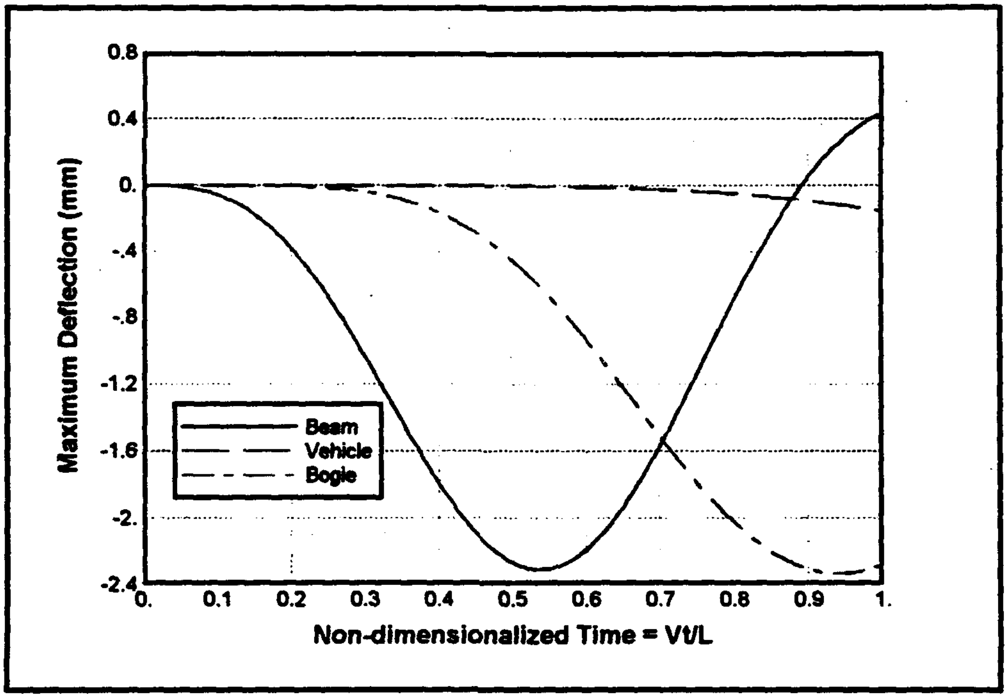


Figure 16. System response for passage speed of 140 mps

The effect of span-crossing velocity on vehicle displacement is shown in Figure 17 (from the Closed-Form Solution method). It indicates that for the system under consideration, lower span-crossing velocities result in greater vehicle movement. However, since all of the vehicles (except for the 30mps

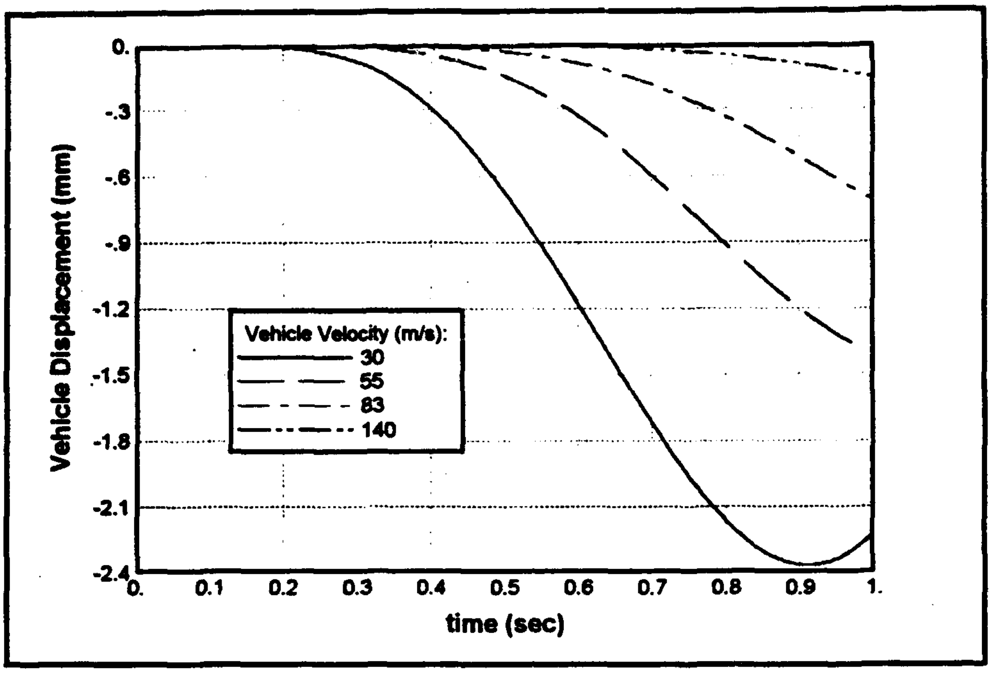


Figure 17. Vehicle displacement as a function of velocity

vehicle) were still moving downward as they reached the end of the span, this trend may not be the case for the next span. While this information is useful for some portions of a vehicle design (such as required suspension clearances), the acceleration experienced by the vehicle is the design variable of greatest impact for determining ride quality of the vehicle.

Unlike the vehicle, the vertical displacements of the bogie(s) in a Maglev system are a very important design criterion. The primary suspension is actually a very narrow magnetic gap between the bogie and the guideway which defines the maximum allowable bogie displacement. If the displacements of a bogie exceed the gap, the bogie will strike the guideway. As a result, the primary suspension of a Maglev system with a small magnetic gap must be very stiff and of relatively low mass in order to closely follow the guideway displacements. Bogie displacement as a function of crossing velocity is shown in Figure 18 for the system under consideration. In each case, the bogie followed the displacements of the guideway (Figure 19) much more closely than did the vehicle (Figure 17) due to its lower mass and much stiffer suspension of the bogie. While all of the bogie displacements were small, they should actually be compared to the corresponding displacement of the guideway in order to determine the minimum allowable magnetic gap for the primary suspension and to ensure against a magnet strike on the guideway. This is demonstrated using Figure 16 (for the 140 mps crossing velocity) and the relationship for the 1st shape given in Equation (22). Figure 16 shows that the maximum bogie displacement (2.34 mm) occurs at a Vt/L of approximately 0.95, which for a 25 m span and V= 140 mps corresponds to t= 0.17 sec and thus a location on the span of x= 23.75 m. According to Equation (1), considering the first mode only with an upward midspan deflection of 0.0003m

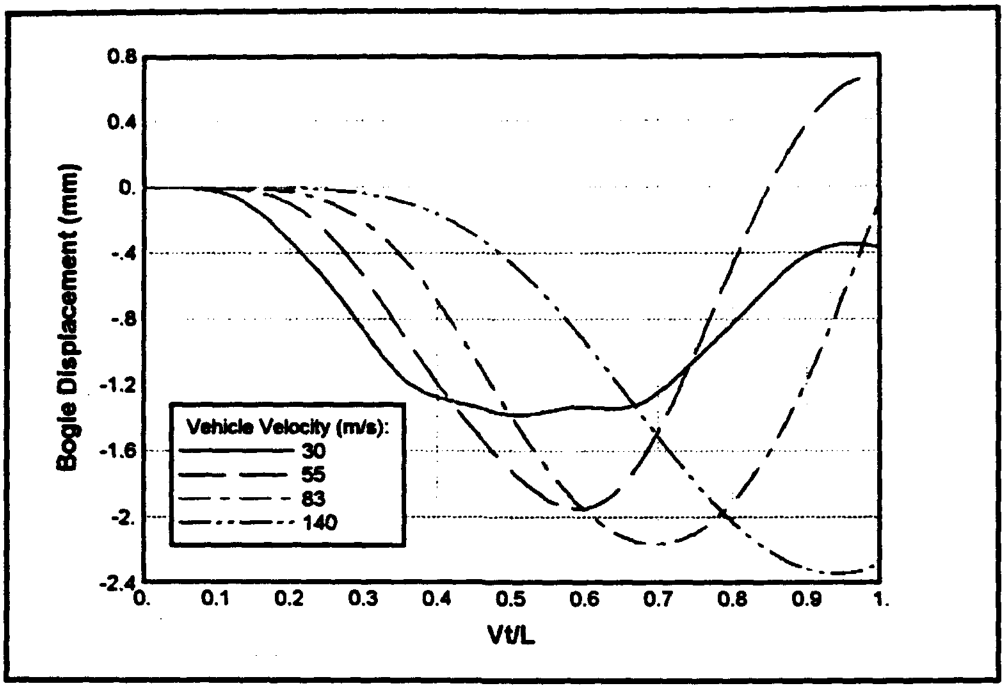


Figure 18. Bogie displacement as a function of velocity

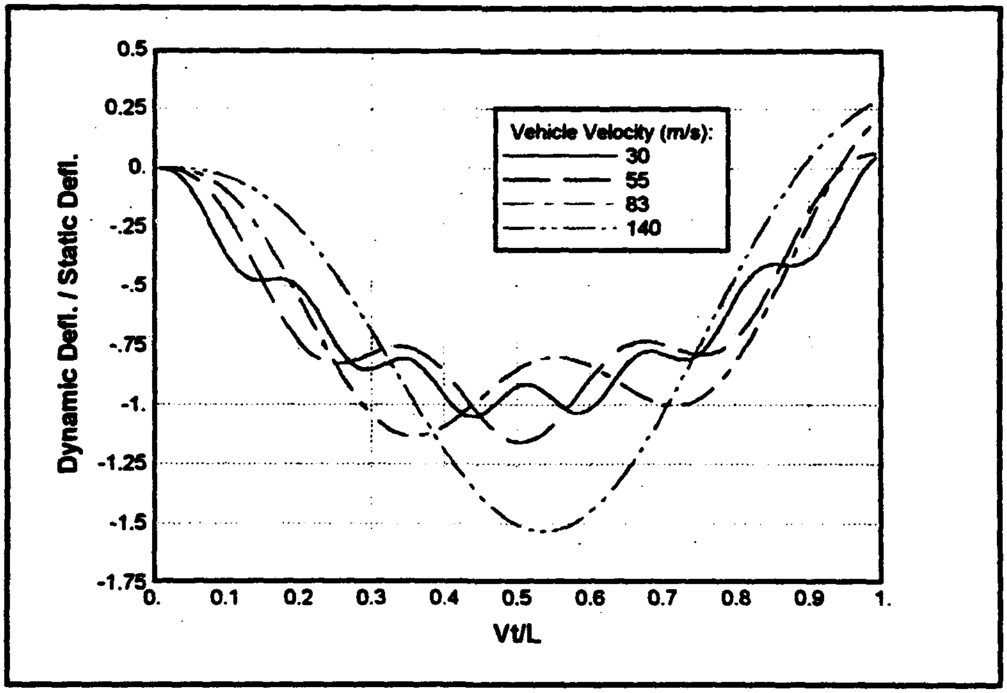


Figure 19. Midspan guideway deflections

at that time, the vertical displacement of the guideway at that location and time will be

$$y_b = 0.0003 \cdot \sin\left(\frac{\pi(23.75)}{25.0}\right) = 0.000016m \cong 0 \tag{27}$$

The minimum allowable magnetic gap will thus be the sum of the bogie and guideway displacement, which in this case will be 2.34 mm + 0 = 2.34 mm.

Figure 19 compares the midspan guideway deflections at different span crossing velocities. To better demonstrate the dynamic effect, the displacements are shown as the ratio of dynamic deflection to the maximum deflection if the load were applied statically (1.51 mm for this case). This ratio is often referred to as the "Dynamic Load Factor" (DLF) and for an elastic structural system, allows the application of static design principles to account for the dynamic response. Note in Figure 19 that the DLF increased as the span-crossing velocity increased. The DLFs ranged from 1.05 at V=30mps to 1.54 at V=140 mps. At a crossing velocity of 140 mps, the span loading frequency will be 140 mps / 25.0 m= 5.6 Hz. This frequency is approaching resonance with the first mode natural frequency of the span of 7.6 Hz, which explains the large increase in deflection observed at that crossing velocity. This points out the fact that resonance situations can be a problem with high velocity vehicles and must be carefully considered in each design. In fact, DLFs greater than 2.0 can easily occur in the case of multiple successive bogies traversing a span at high speeds (Lever 1993).

Although superconducting technology has greatly improved the capabilities of magnets, required forces from the magnets must still be kept within specific design limits. Otherwise, the gap limits may be exceeded, resulting in a magnet strike on the guideway, or even possibly a vehicle separation from the guideway (in the case of a "repulsive" type of magnetic suspension). A plot of the primary suspension force variation (which represents the magnet forces) for the system under consideration is shown in Figure 20 for varied span crossing velocities. A maximum force variation of only approximately 2.1 percent occurred with the 140 mps crossing velocity. Since the force variations were so small, a comparative FE analysis was conducted where a constant unsprung force was moved across the same FE beam model. The force was equivalent to the combined static weight of the sprung masses in the previous analytical cases. As expected, the beam response was almost identical to that where the sprung mass was used, showing that the effect of VGI was minimal for this case. Again however, the VGI effect would likely be greater if more than one span were considered in the analysis and VGI should never be ignored until comparative analyses are conducted.

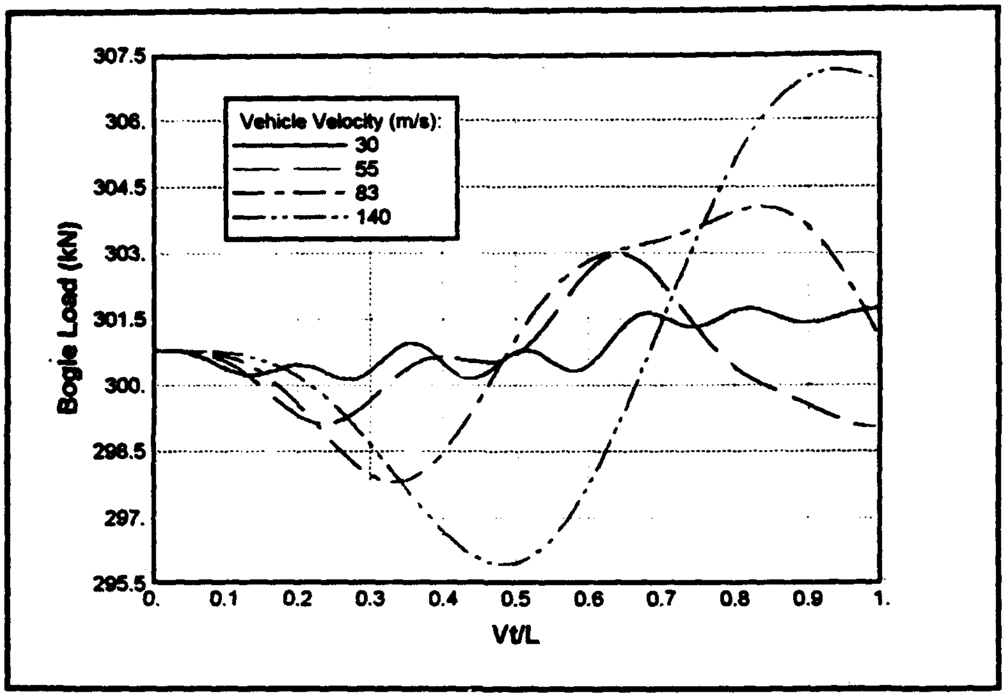


Figure 20. Variation of bogie loads with vehicle velocity

4 VGI Model Application

Introduction

The VGI Model was used to analyze the Foster-Miller SCD (Foster-Miller 1993). This was done in order to demonstrate the use of the VGI Model and to verify its applicability to actual Maglev systems. The Foster-Miller SCD was chosen for this demonstration because it offered both a challenging and yet somewhat generic application of the VGI Model. It is important to note that this analysis was done only for demonstration purposes and its results should not be used for an actual assessment of the Foster-Miller SCD.

The Foster-Miller concept is an EDS (Electro-Dynamic System) generally similar to the prototype Japanese MLU002. Superconducting vehicle magnets generate lift by interacting with null-flux levitation coils located in the sidewalls of a U-shaped guideway. Similar interaction with series-coupled propulsion coils provides null-flux guidance. Its unique propulsion scheme is called a locally commutated linear synchronous motor (LCLSM). Figure 21 shows an overall view of the concept.

The baseline Foster-Miller vehicle consists of two 75-passenger modules with attached nose and tail sections. Fabrication of smaller or larger vehicles is possible by incorporating fewer or additional passenger modules. The two-car baseline vehicle was used for the demonstration herein since prior analyses by Foster-Miller (Foster-Miller 1993) showed this combination to produce the worst-case loading on the guideway.

The Foster-Miller guideway concept is shown in Figure 22. The guideway superstructure is a unique open-cell, integral sidewall structure constructed from modular units. Two symmetric halves are coupled together by a series of intermittently spaced truss-type diaphragms. The units are held together by posttensioning tendons that run transversely through the sections. The outer beam portions are reinforced longitudinally by a combination of pre- and posttensioned steel tendons in the lower half and FRP (Fiber Reinforced Plastic) tendons in the upper half. The girders are erected on their pier supports as simply-supported spans and then made two-span continuous with the application of external FRP posttensioning at every other support.

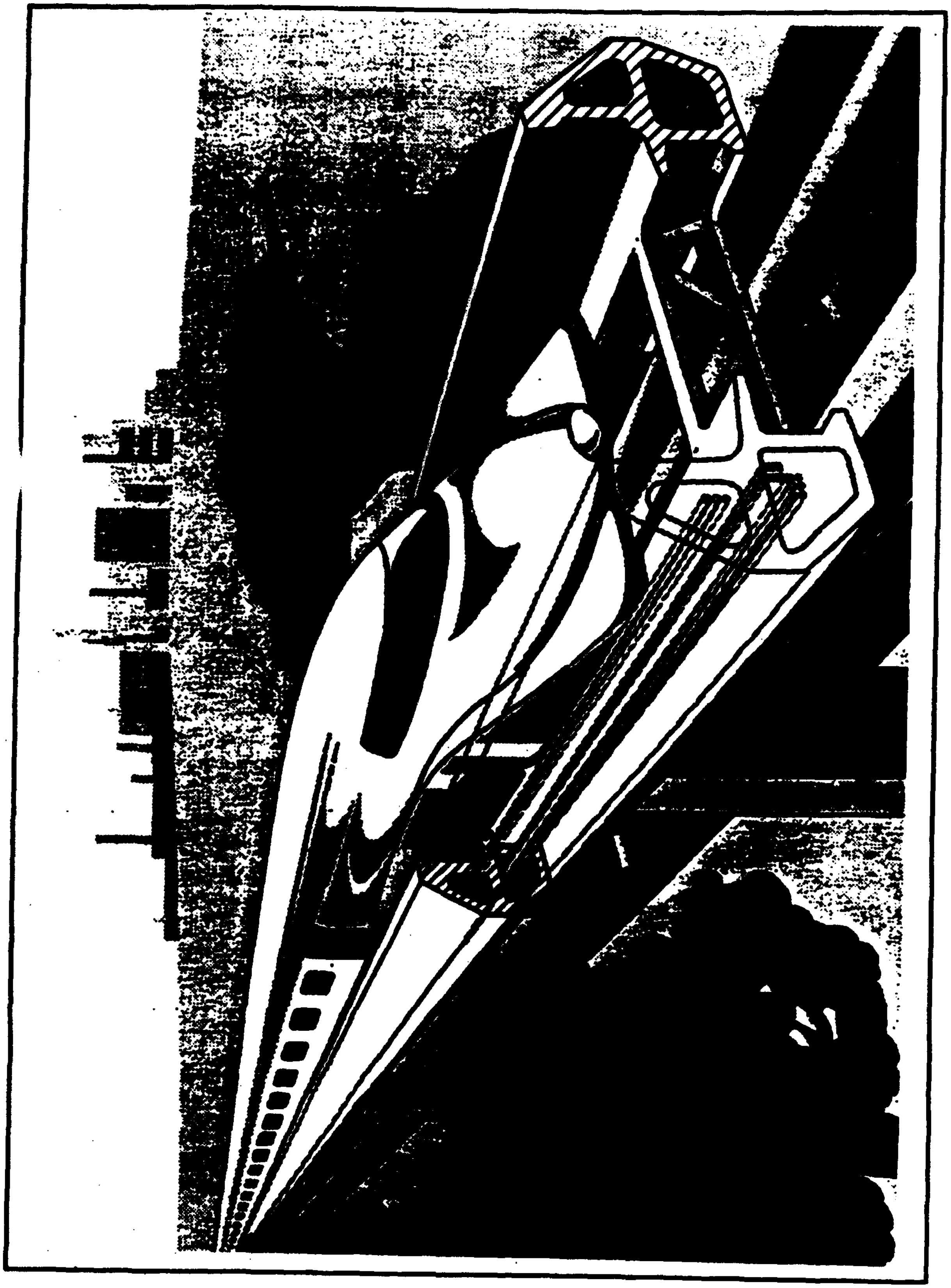


Figure 21. Foster-Miller Maglev System Concept

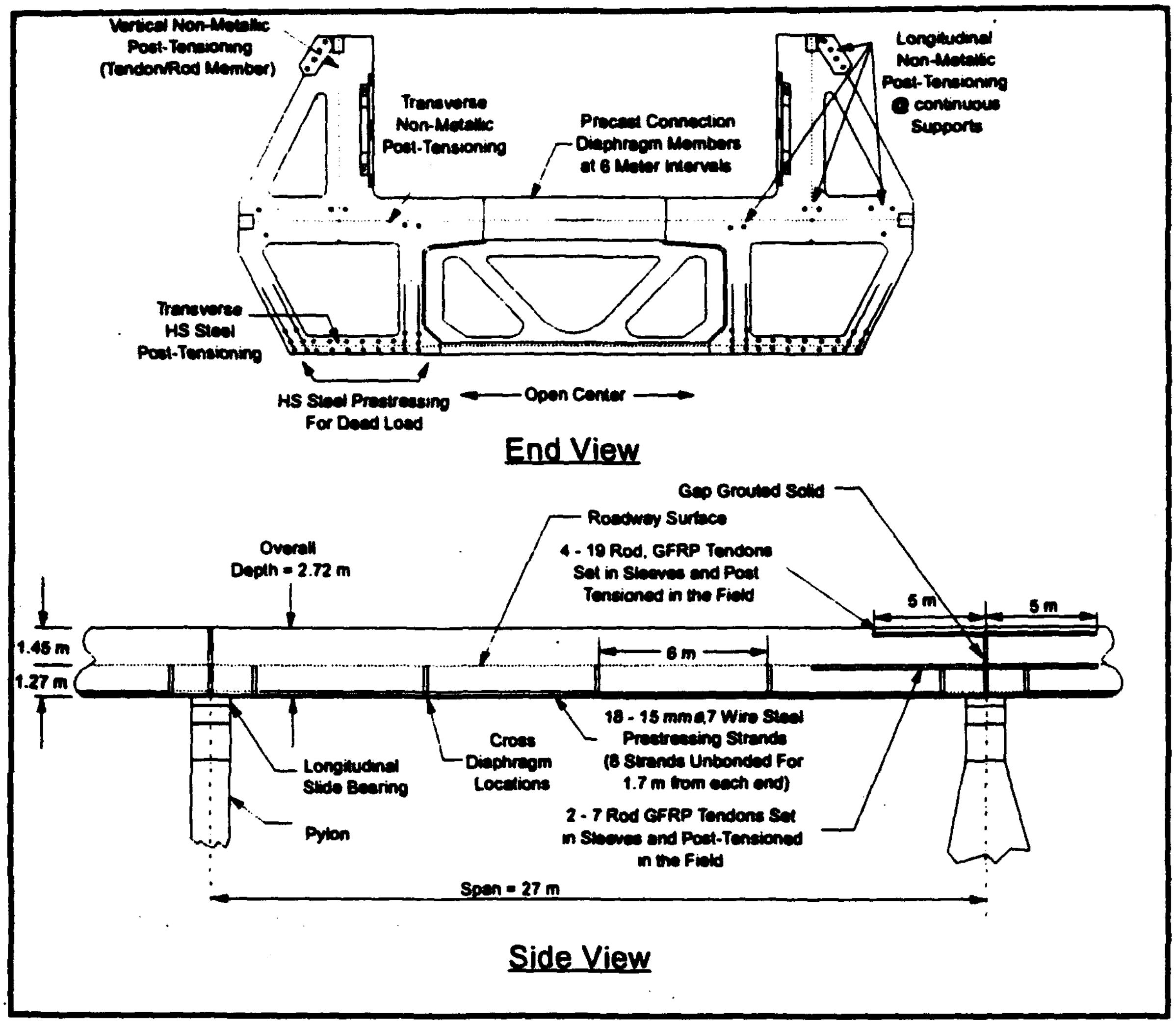


Figure 22. Foster-Miller guideway concept

FE Mesh Creation

The first step in the VGI Model is to generate the FE mesh of the guideway. Several limiting criteria were discussed in Chapter 2 for the mesh refinement. The first criteria serves to insure that the mesh is refined enough to accurately represent all bending modes that may be excited by the vehicular loading frequencies, which are mainly functions of vehicle speed and bogie spacing. According to the criteria of Chapter 2, if only guideway deflections were of concern, only the first two guideway modes would have been necessary to represent. However, since a structural analysis is part of the VGI Model (i.e. bending moments and stresses) and the guideway is two-span continuous, the finite elements for this mesh had to be short enough (along the length of the guideway) to accurately represent the first six (i.e. *3k*, three times

the number of spans) vertical guideway bending modes. Only vertical bending modes were required since only vertical loadings were applied. If horizontal loadings were also applied, the same criteria should also be applied in deciding how many of those modes to represent.

The first six general bending modes for a two-span continuous beam are shown in Figure 23. Approximately 15 to 20 elements per span would be required to adequately represent all of these modes. Using 20 elements, the maximum element length would be 27m/20= 1.4 meters.

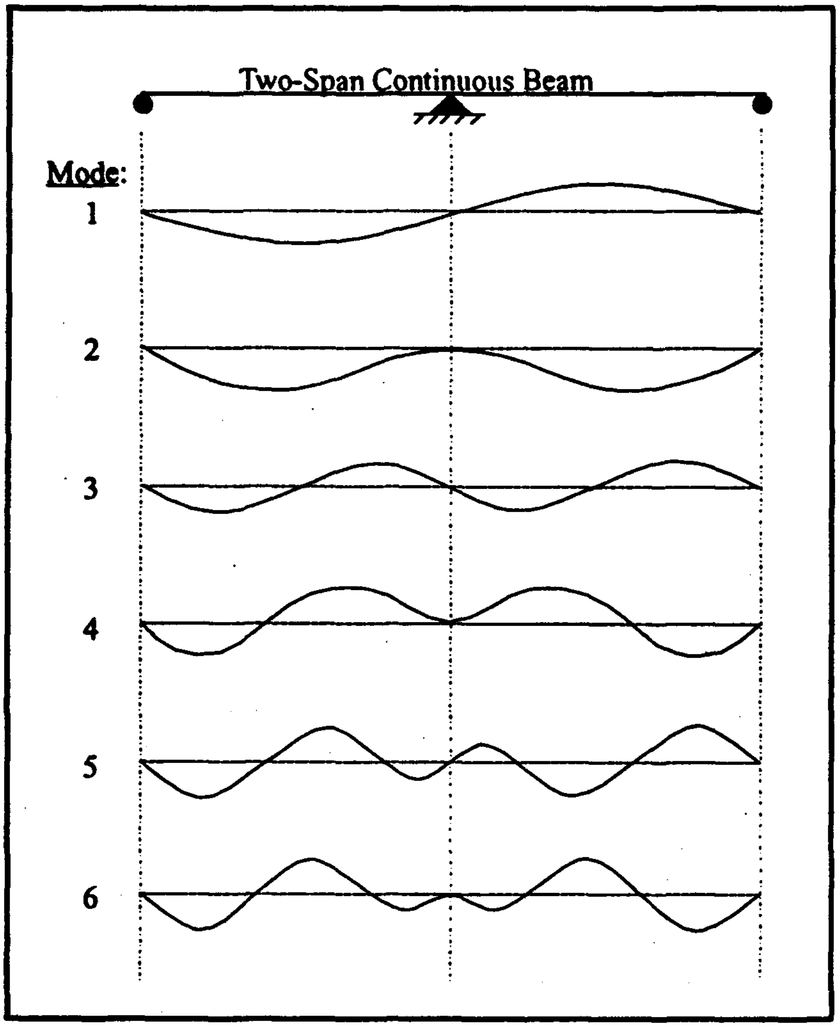


Figure 23. First six bending modes for a two-span continuous beam

Although the above generic criteria should usually be sufficient to insure accuracy of the mesh, it is also prudent to specifically compare the guideway loading frequency to the modal frequencies as follows: The Foster-Miller vehicle is depicted in Figure 24. Its bogies are approximately 24m on center. For a vehicle velocity of 140m/s (the maximum requirement for the SCDs), the guideway loading frequency will be approximately 140m/s divided

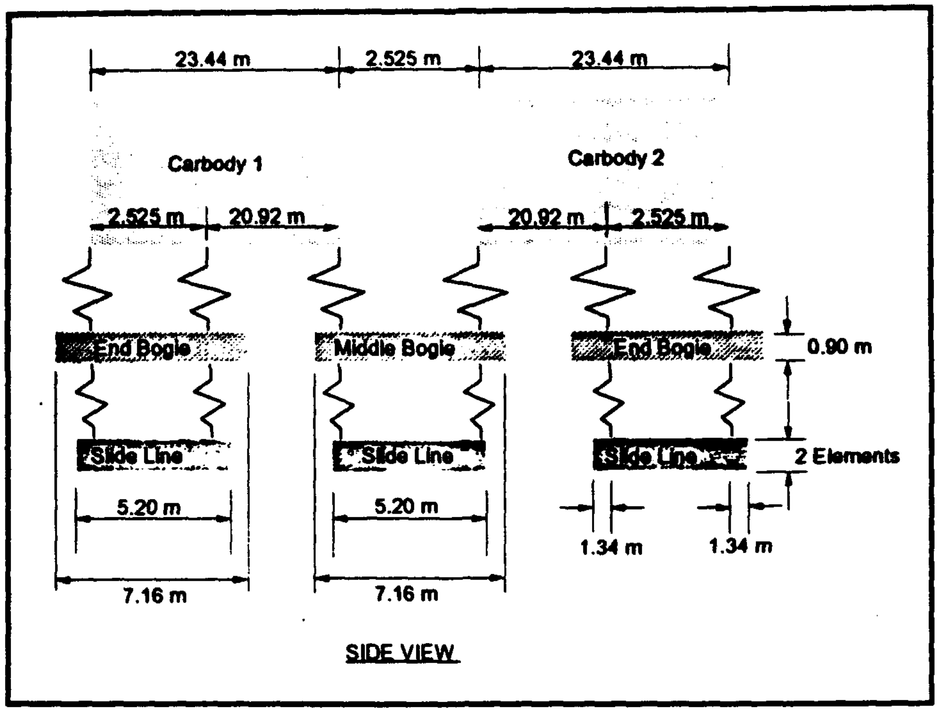


Figure 24. Foster-Miller SCD Vehicle

by 24m, which equals to 5.8 Hertz. This frequency is very close to the first and second bending mode frequencies for the guideway (calculated based on the guideway bending stiffness and span length) and thus will primarily excite only these modes. Therefore, the above general requirement of a mesh to represent the first six bending modes was sufficient.

The other VGI-specific criterion for the FE mesh is that the loaded elements must be less than two times the shortest bogie length. The vehicle bogies in Figure 24 are all 5.2m in length. Therefore, the element lengths must be less than (5.2)(2)= 10.4m. The above criteria require much shorter lengths than this and thus will control the mesh size.

The FE mesh of the guideway is shown in Figure 25. Only the superstructure of the guideway was modeled since the substructure response will generally be of such low frequency that it will not affect the vehicular ride quality. The lengths of the elements (longitudinally along the guideway) were 0.5 m, much smaller than actually required by the above criteria. This mesh was actually one from a previous set of analyses (Lever 1993) in which localized bending stresses in the sidewalls were studied, which required a more refined mesh. While a more refined mesh will only improve the accuracy of the VGI solution, it unnecessarily increased the size and complexity of the FE solution and thus the required computer CPU time.

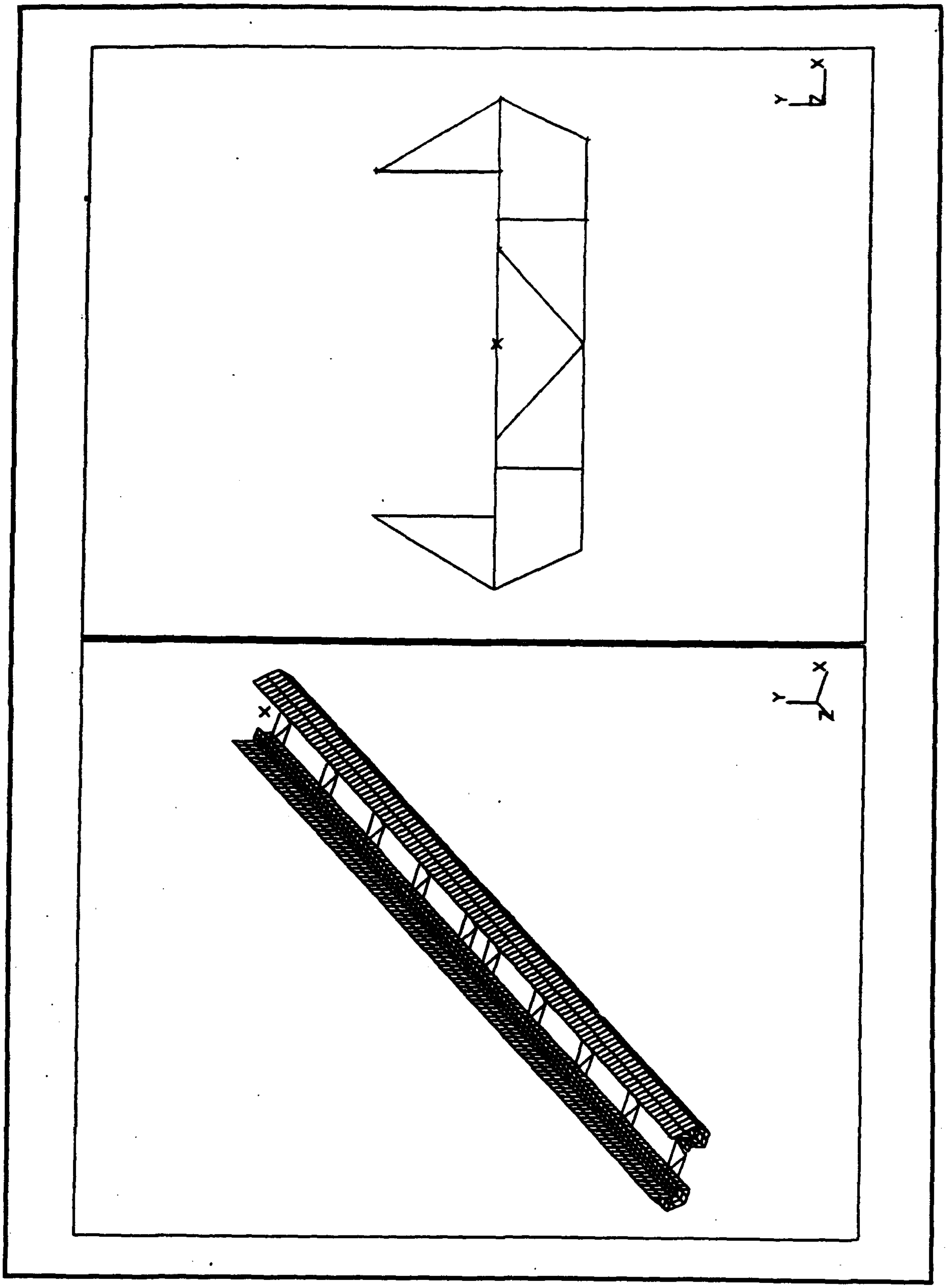


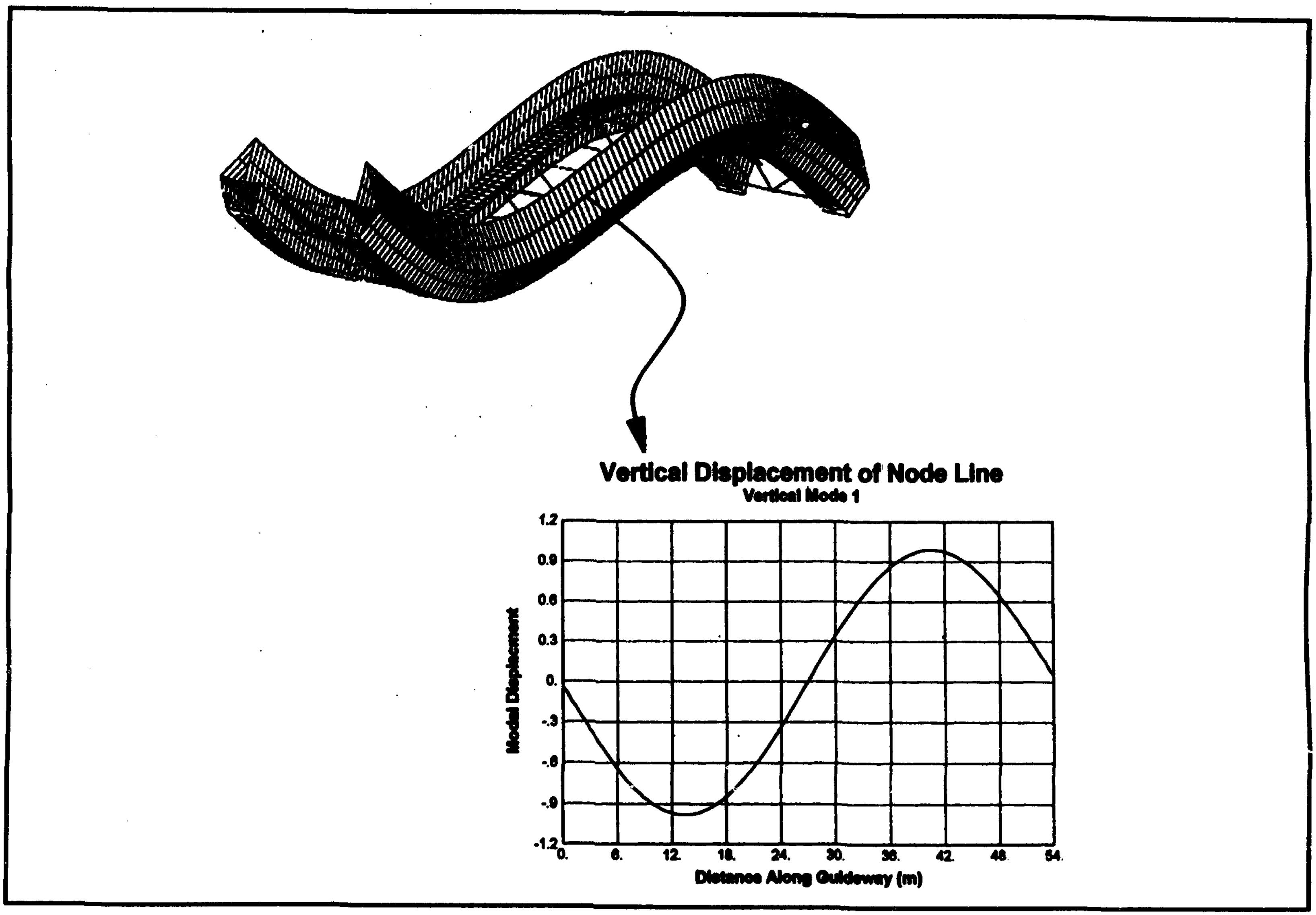
Figure 25. Finite element mesh of Foster-Miller guideway

Modal Analysis

Once the FE mesh of the guideway was constructed, the "FREQUENCY" analysis option of the ABAQUS FE code was used to obtain all of the necessary mode shapes and modal parameters. The deflected shapes for the first four vertical modes are shown in Figure 26. Once these results were obtained, the deflected shapes of the first six modes were reviewed to insure that the mesh was sufficiently refined to accurately represent their shape. In this case, the smooth deflected shapes indicated that the mesh refinement was more than sufficient.

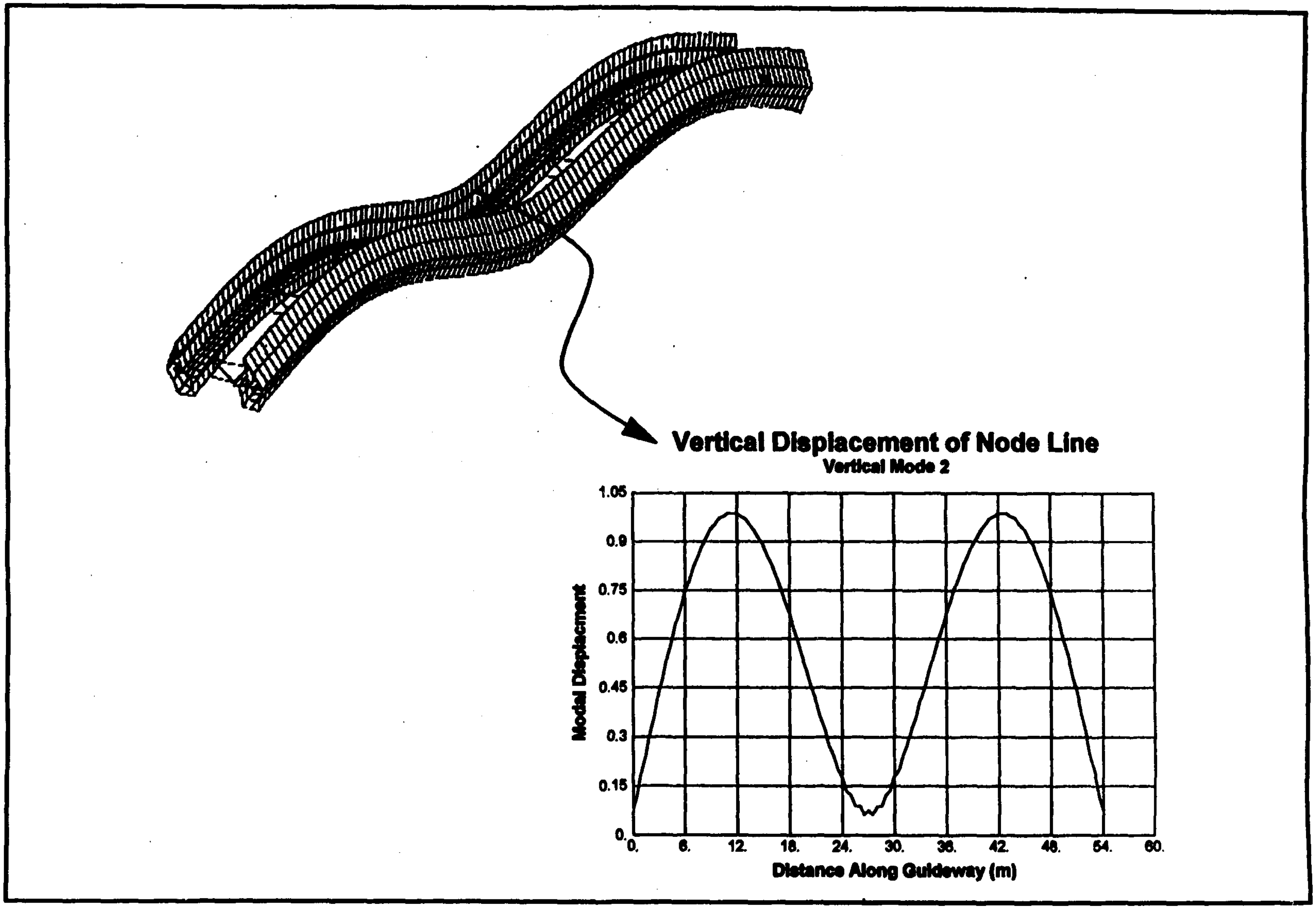
Certain of the mode shapes were then chosen for use in the RQAC. As discussed in Chapter 2, only those with frequencies less than three times the vehicular crossing frequency should be used. These are the only ones which significantly contribute to the vehicular response. In this case, the vehicular crossing frequency was approximately 5 Hz. Thus, all vertical bending modes with frequencies less than 15 Hz were chosen. For the Foster-Miller guideway, this corresponded to the first four modes shown in Figure 26. They had modal frequencies of 5.7, 8.5, 8.5, and 10.3 Hz, respectively. Note that the first two modes were of conventional shapes for two-span continuous beams (refer to Figure 23). Yet, the third and fourth modes were unique to the Foster-Miller concept due to its modular construction. They were basically the first and second bending modes (Figures 26a and 26b) acting out of phase on opposite sides of the guideway. These modes will significantly contribute to vehicular roll motion. These unconventional mode shapes emphasize the importance of 3-d analyses of complex structures. Modal Participation Factors (also part of the ABAQUS FREQUENCY analysis output) could also have been used to determine modes of importance for use in the RQAC.

The 3-d mode shapes of Figure 26 were reduced to a 2-d form as required for use in the RQAC. Since the vehicle only contacts the guideway along specific lines (i.e. along the vertical sidewalls on each side), the displacements along these lines were all that were actually required. In this case, only the vertical VGI response was being considered. Therefore, only the vertical deflections along the lines were required. The 2-d displacements of the mode shapes are also shown in Figure 26. Note that only one set of 2-d data is shown for each mode shape. For use in the RQAC, this data would actually be applied to both sides of the vehicle (i.e. two magnet lines) in order to excite the vehicle's roll response. It is important to note here that the RQAC portion of the VGI Model was not used in this example and thus this portion of the exercise only served to demonstrate how the 2-d data would be supplied to the RQAC in the event of its use.



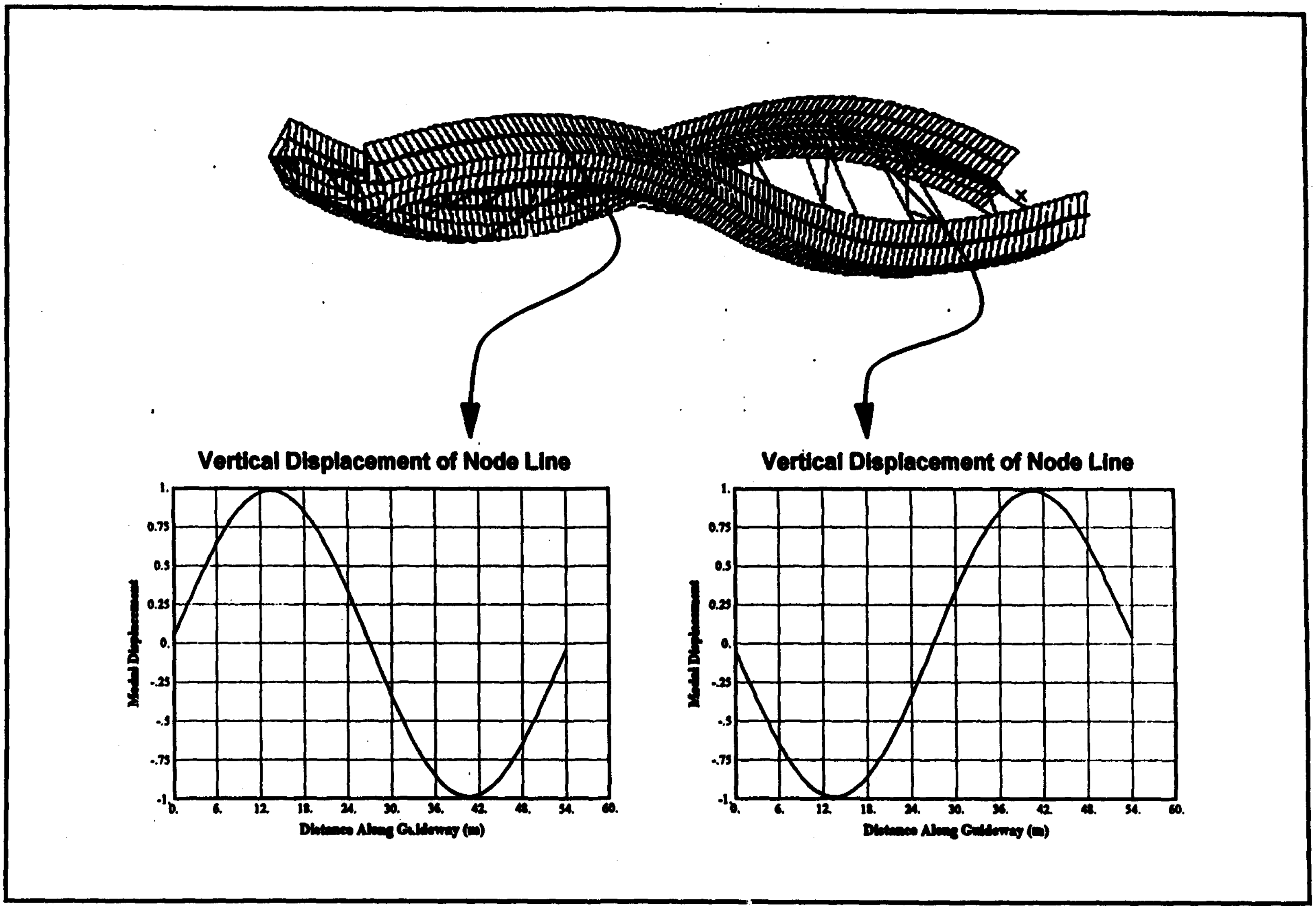
a. First vertical bending mode

Figure 26. Mode shapes for Foster-Miller guideway (Continued)



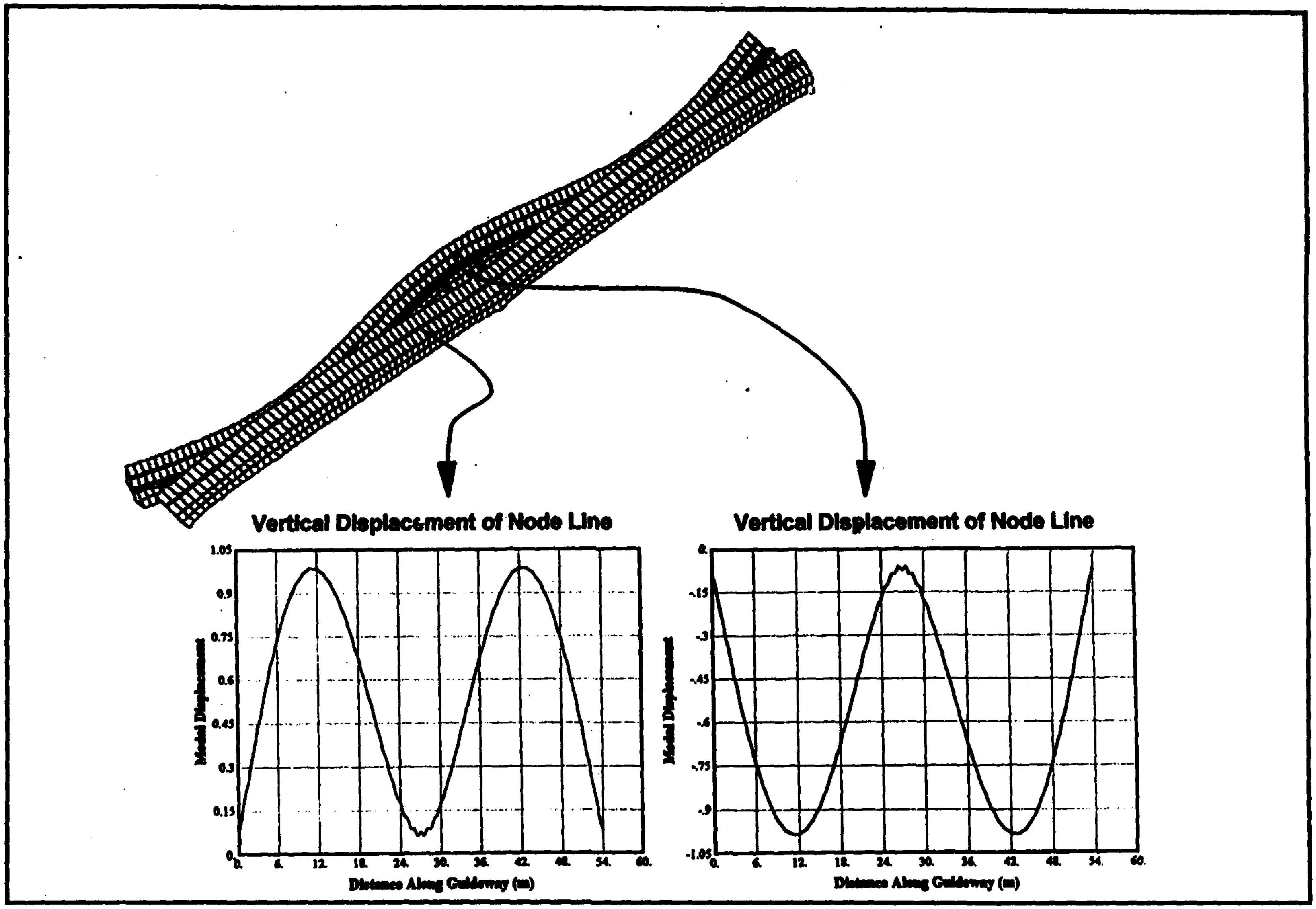
b. Second vertical bending mode

Figure 26. Mode shapes for Foster-Miller guideway (Continued)



c. Third vertical bending mode

Figure 26. Mode shapes for Foster-Miller guideway (Continued)



d. Fourth vertical bending mode

Figure 26. Mode shapes for Foster-Miller guideway (Concluded)

Vehicular Loads

Modifications to the RQAC by the VNTSC were not completed at the time of this report. Therefore, the vehicular bogie load output from the RQAC, required for input to the GAC, was manually generated in a simplified form. The resulting analysis was equivalent to the propagation of the static forces due to the distributed vehicle weight across the guideway.

The two-vehicle consist as analyzed was previously shown in Figure 24. The vehicle weights were distributed out evenly to the bogies, resulting in the forces at the guideway level of 70.5 kN for the front and rear bogies and 129 kN for the middle bogie. These forces were used in the BOGLOAD.DAT file shown in Appendix B. It is important to note that this file would have been much larger if the time-varying bogie loads from the RQAC had been available.

Structural Analysis

The "DYNAMIC" analysis option was used to perform the dynamic analysis of the FE mesh shown in Figure 25. The loadings from the BOGLOAD.DAT file were applied to the loaded element set (ELSET) through use of the DLOAD subroutine. The pertinent portions of the ABAQUS input file are shown in Appendix B. The vehicle input data stored in the VEHINPUT.DAT file is also shown in Appendix B. This file was built automatically through execution of the DLINGEN.FOR program (Refer to Appendix A).

As discussed in Chapter 2, a small analysis timestep was required in order to insure that loads were applied to each of the integration points within the loaded elements. Since the integration point spacing was approximately 0.25m and the vehicle velocity was 140m/s, the maximum timestep was chosen as 0.25/140= 0.002. The analysis was run for a total time of 0.76 seconds in order for the 52-meter long vehicle (traveling at 140m/s) to get completely onto and off of the 54-long guideway. With 0.002 second timesteps, this corresponded to a total of 380 analysis increments.

The resulting midspan displacements of the guideway for spans 1 and 2 are shown in Figure 27 as a function of the location of the front of the vehicle (i.e. head position). These results compared very closely to those from a previous dynamic analysis of this system using a somewhat less-refined analytical technique (Lever 1993). Although not accomplished in this example, the analysis output could also include guideway accelerations, stresses, strains, and reaction forces.

The final step in the VGI Model application is to verify the overall solution accuracy by comparing the guideway deflections predicted by the RQAC and GAC. However, since the RQAC was not used in this exercise, this could not be accomplished.

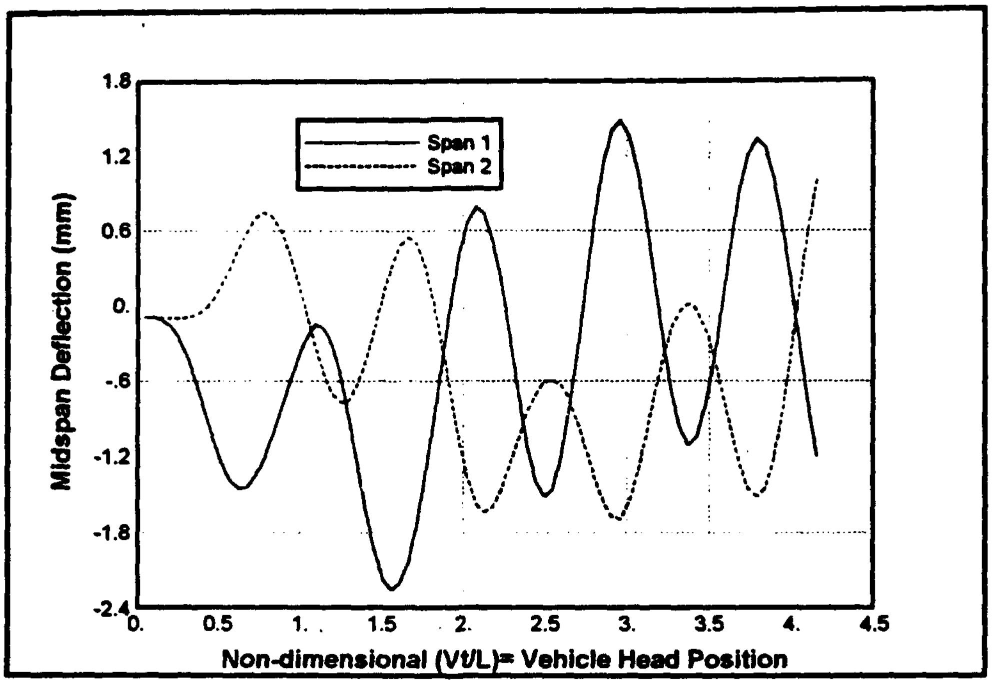


Figure 27. Midspan displacements for spans 1 and 2 of Foster-Miller guideway

5 Conclusions and Recommendations

Conclusions

A VGI Model was developed between the WES and the VNTSC. The Model consists of a methodology and toolkit for the linkage of two separate analytical codes, the RQAC and the GAC. By making the input/output from each of these codes interactive, the vehicle-guideway interaction process is taken into account in both analyses. The Model has two distinct applications: It can be used to predict the vehicle ride quality to be expected from a given vehicle and guideway design; and to accurately predict the dynamic deflections and stresses experienced throughout a complex guideway structure as a result of a vehicle passage.

The Model was verified and validated successfully in Chapter 3. The results from all three VGI analytical methods (closed-form solution, FE solution, and the VGI Model) compared very closely. The structural response predicted by the VGI Model lagged slightly behind that from the other analyses. This was attributed to the manner in which the integration points within the finite elements are loaded by the DLOAD routine and the fact that no rise or decay time is used for the loadings. Although this causes a slight lag in the response of the structure, it only amounts to a very small error and thus should produce no real problem in an actual VGI analysis.

The RQAC was not discussed in detail in this report. However, it was demonstrated in Chapter 2 that the solution approach used in the RQAC is valid for the evaluation of ride comfort and the determination of the low frequency (less than about 15Hz) variations in the vertical forces acting between the vehicle and guideway. Specific criteria for vehicle/guideway combinations were given for which the RQAC approach is valid. The current German TR07 Maglev system and the four U.S. Maglev SCDs were compared and found to meet these criteria. For vehicle/guideway combinations that do not meet these criteria, a different numerical solution technique may be required.

Ride quality requirements for Maglev vehicles will generally severely limit vehicle movements and thus the variation in vehicular forces at the guideway level. As a result, a complete VGI analysis may not always be

necessary for guideway analysis/design considerations. In many cases, movement of static vehicular gravity loadings across the guideway mesh, as done in Chapter 4, will likely be sufficient for all guideway concerns. However, this assumption must be used with great care and should only be applied after careful review of the specific system under consideration.

Recommendations

Because all Maglev systems are unique and present different analytical concerns, application of the VGI Model to specific problems must be done with great care, and only by personnel experienced in its use. In order to gain experience in the use of this Model, future work should include its application to specific Maglev systems, such as the existing German and Japanese systems and the current U.S. Maglev concepts. However, this work should only be accomplished after completion of the RQAC development (currently underway) when the entire Model can be utilized.

The VGI Model is currently very "code specific" in that it was developed specifically around the ABAQUS FE code and the VNTSC vehicle dynamics code. As a result, it will have limited use to those without access to and experience with these codes. However, with some effort, the Model could be adapted to any structural and vehicle dynamics codes. Future work in this area should be directed toward the development of a more generic, widely usable VGI Model. The FE method discussed in Chapter 3 showed great promise in this respect. This method would allow a single FE analysis code, such as ABAQUS, to be used for the entire VGI analysis; i.e. both the vehicle and the guideway. It would also allow for the consideration of parameters which are not available in the current VGI Model such as curved guideways, wind loadings on the vehicle, and flexible vehicles.

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Appendix A
Fortran Source Code for Use of DLOAD

File "DLINGEN.FOR" is the first file accessed by the user. It prompts the user for the appropriate input and then builds an input file which is used to provide input directly to the "DLOAD.F" subroutine during execution of the ABAQUS FE analysis program. The file which contains the actual bogie force-time histories must be entitled "BOGLOAD.DAT"

Program DLINGEN.FOR

```
c*****************************************************************
c
C    This file will create an input deck for the DLOAD subroutine
c
c*****************************************************************
c
      CHARACTER *72 OUTFILE
      INTEGER   *4  NUMGRP
      REAL    *8  BOGLNTH,BOGHT,THICK,V,NORMAL,ZF(100),ZB(100)
c
c
c     Read bogie information
c
c
      write(*,*) 'This program creates an input deck for the DLOAD Sub'
      write(*,*) 'Enter the name of the output File for DLOAD:'
      read(*,'(A30)') outfile
      open(unit=61,file=outfile,status='unknown')
c
      write(*,*) ' Enter bogie length (meter):'
      read(*,*) boglnth
```

```
      write(*,*) ' Enter bogie height in meter:'
      read(*,*) boght
c
      write(*,*) ' Enter vehicle speed in mps:'
      read(*,*) v
c
      write(*,*) ' Enter element thickness in m:'
      write(*.*) ' Element thickness only important for shear type loadings
     &where the pressure is distributed over the volume of  the element'
      read(*,*) thick
c
      write(*,*) ' Enter number of group of bogies:'
      read(*,*) numgrp
c
       write(*,*) ' Enter "1" if vertical vehicle loads are applied Normal to
     &loaded elements, Enter "2" if applied in Shear:'
       read(*,*) normal

c     Read the front and back coordinates for each bogie
c
       Do I=1,numgrp
         write(*,*)'Enter z-cord for the', I,'th front magnet group:'
         read(*,*) zf(i)
c
         write(*,*) 'Enter z-cord for the', I,'th back magnet group:'
         read(*,*) zb(i)
       end Do
c
c
         write(61,*) boglnth,boght,v,thick,numgrp
c
         Do i=1,numgrp
           write(61,*)zf(i),zb(i)
         end Do
c
c     to compile fortran file do the following:
c
c     create a source code, for example the name of this file is conv.f
c
c     type:cf77 -o name conv.f  <--------to compile conv fortran source code
c
c     execute the fortran file by typing:name <----name of the compiled file
c
       stop
       end
```

Subroutine DLOAD.F

```
    SUBROUTINE DLOAD
(F,KSTEP,KINC,TIME,NOEL,NPT,COORDS,JLTYP)
C
     INCLUDE 'ABA_PARAM.INC'
C
     DIMENSION TIME(2), COORDS(3)
C
c*****************************************************************
c
C   This routine assumes ABAQUS units are meters
c
C    This file will be called by the maglev FE input deck for each
c   loaded intergration point and at each increment of time step.
c
c   NOTE: external file created by 'DLINGEN.F' will
c        be called in by this subroutine.  It can have any name, but is named
         "VEHINPUT.DAT" herein.
c
c*****************************************************************
c
c   CHARACTER *72 OUTFILE
    INTEGER  *4  NUMGRP, NUMROW
    REAL  *8  BOGLNTH, BOGHT, V, THICK,NORMAL,
ZF(100),ZB(100),
   +READAR(2,81), DELF
c
c   'BOGLOAD.DAT' contains the bogie load-time history data with the
following format:
c          first line:       n  ----number of data points
c          successive lines: (4*numgrp+1) columns
c
c    The file generated by 'DLINGEN.F' (called VEHINPUT.DAT herein)
contains the
       vehicle information in the following format:
c                1st line:  boglength,boght,v,numgrp,normal
c                following lines: zb(i),zf(i) ---i from 1 to numgrp
c
    OPEN(unit=61,file='/BOGLOAD.DAT',status='old')
    OPEN(unit=62,file='/VEHINPUT.DAT',status='old')
c
c   Read bogie information fron input file unit 62
c
     READ(62,*) boglnth,boght,v,thick,numgrp,normal
c
```

```
        Do i=1,numgrp
          read(62,*)zf(i),zb(i)
        end Do
C*********************************************
C end of definable parameters
C *********************************************
C
C     Zero out array 'READAR'
c
      J= (NumGrp * 4 + 1)
c
      Do  i= 1,J
        READAR(1,i)= 0
        READAR(2,i)= 0
      end Do
C
C
****************************************************************
**********
C
      F = 0
C start w/ F=0 for case of no bogie over the integ. point
C
C Now, w/ the current integration point defined by COORDS(3), check to see
if
C  any bogies are over it
C
      iGroup = 0
C stays 0 unless a bogie is found over the integ. point
C
      Do 100 I = 1,NumGrp
        IF (((V * TIME(1) - zf(i)) .GE. COORDS(3))
     +.AND. ( (V * TIME(1) - zb(i)) .LE. COORDS(3))) then
            iGroup = i
            go to 6
          END IF
c
100       continue
c
         GO TO 10
C    i.e. If no bogie over the point, go to end
c
6      CONTINUE
C
C   The following is VERY dependent upon proper order of Bogie load input
c   file and the definition of  the variable NORMAL
c
        If (JLTYP .EQ. 2) GO TO 70
```

```
c     JLTYP=2 corresponds to body force BYNU
c
        If (JLTYP .EQ. 20) GO TO 80
c     JLTYP=20 corresponds to normal force PNU
c
c         Write(*,*) 'wrong JLTYP, must be 20 or 2'
c
70      If (NORMAL .EQ. 1) GO TO 81
         If (NORMAL .EQ. 2) GO TO 71
c
80      If (NORMAL .EQ. 1) GO TO 71
         If (NORMAL .EQ. 2) GO TO 81
c
71       If (COORDS(1) .GE. 0) NUMCOL = (igroup*4)-2
        If (COORDS(1) .LT. 0) NUMCOL = (igroup*4)
        GO TO 91
C
81       If (COORDS(1) .GE. 0) NUMCOL = (igroup*4)-1
        If (COORDS(1) .LT. 0) NUMCOL = (igroup*4)+1
C
91       CONTINUE
C
C
*****************************************************************
***
C
C  Now we know from which NumCol in the BOGLOAD.DAT file to read the
Force.  We
C  must now perform an algorithm to read thru the load file and find the TIME
C    of interest (i.e. appropriate row).   We will actually bracket 2 time values
C   that are above and below the time of interest, and interpolate a FORCE
     value
C
C        Create a Matrix consists of time and wheel load corresponding to
           numcol.
c
c       NOTE: Numcol identifies the column corresponding to position (left or
          right)
c       and direction (shear or normal) of the bogie load.
c
c      NumRow is the number of rows in the BOGLOAD.DAT file
c      READAR(j,1) ---------time increment column
c      READAR(j,NUMCOL)------required bogie load column
c
        READ(61,*) numrow
      DO J=1,numrow
         READ(61,*) (READAR(2,KK),KK=1,NUMCOL)
        IF (TIME(1) .EQ. READAR(2,1)) THEN
```

```
          F=READAR(2,NUMCOL)
          GO TO 50
        END IF
c
        IF (TIME(1) .GT. READAR(2,1)) THEN  GO TO 11
           ELSE GO TO 20
c       END IF
c         NOTE: Change the Value of F when JLTYPE .EQ. 2 (shear type
loading on the element)
c
11        CONTINUE
c
        DO KK=1,NUMCOL
          READAR(1,KK)=READAR(2,KK)
        END DO
c
      END DO
c
C       The time corresponding to jth row of input file
c        is less than the current value of time step.
c        Hence, the current force is in between jth and
c        (j-1) th row of input file. Where, timerow has the jth value.
C
C      Interpolate between 2 lines of input to find specific
c        magnet force at specific MAGtime.
c
20     Fdif= (READAR(2,numcol) - READAR(1,numcol))
      Tdif=(READAR(2,1)-READAR(1,1))
      delF=Fdif/Tdif*(TIME(1)-READAR(1,1))
c
      IF(delf .GE. 0.) then
        F=READAR(1,numcol)+delf
       ELSE
        F=READAR(2,numcol)+delf
       END IF
c
C      Next, get F in terms of pressure
c
50          IF(JLTYP .EQ. 20) THEN
               F=F/(BOGlnth*BOGHT)
            else
               F=F/(BOGlnth*BOGHT*THICK)
            END IF
C      Subroutine Complete
10    CLOSE(unit=61,status='keep')
       CLOSE(unit=62,status='keep')
      Return
      End
```

Appendix B
ABAQUS Input Files

Note: This file is the specific input file used for the slab analysis using the VGI Model as discussed in Chapter 3:

```
*HEADING
 SIMULATION OF LOAD-INPUT FOR A MAGLEV SYSTEM
***   An input file to illustrate the use of DLOAD external subroutine
***   in an ABAQUS file.
**
****************************************************************
*******
*NODE
1,-2,0,0
51,-2,0,25.
2001 , 2,0,0.
2051, 2. , 0.,25.
*NGEN,NSET=LEFT
1,  2001 , 250
*NGEN,NSET=RIGHT
51, 2051, 250
*NFILL
LEFT,RIGHT,50,1
**
*NSET, NSET=FIX, GENERATE
1,  51, 1
251, 301, 1
2001,2051, 1
1751, 1801, 1
*ELEMENT,TYPE=S8R5
1,  1,3,503,501,  2,253,502,251
*ELGEN,ELSET=SLAB
1,4,500,100,25,2,1
```

```
*ELSET,ELSET=LWHELL,GENERATE
1,25,1
*ELSET,ELSET=RWHELL,GENERATE
301,325,1
*MATERIAL,NAME=M_GUIDE
*ELASTIC
30E9 , 0.3
*DENSITY
954.72
**2400.00 for beam element
*SHELL SECTION,MATERIAL=M_GUIDE,ELSET=SLAB
1.8647, 7
*BOUNDARY
LEFT , 1,3
RIGHT, 1,2
**
*USER SUBROUTINES,INPUT=dload.f
**
*STEP,INC=330
LOAD INPUT
*DYNAMIC,DIRECT, NOHAF
.005,0.30
*DLOAD,OP=NEW
 LWHELL,PNU
 RWHELL,PNU
**
*NODE FILE
U
*END STEP
```

<u>Bogie Load Data File, BOGLOAD.DAT,</u>
<u>As Used For VGI Analysis of Slab in Chapter 3</u>

Note: This file actually has 201 lines of data. However, only one page is shown in the interest of space:

201				
0	35350	0	35350	0
0.001506	35349.3198208	0	35349.3198208	0
0.003012	35347.2858233	0	35347.2858233	0
0.004518	35343.9175744	0	35343.9175744	0
0.006024	35339.247506	0	35339.247506	0
0.00753	35333.3206484	0	35333.3206484	0
0.009036	35326.1942585	0	35326.1942585	0
0.010542	35317.9373476	0	35317.9373476	0
0.012048	35308.6301111	0	35308.6301111	0
0.013554	35298.3632658	0	35298.3632658	0
0.01506	35287.2372984	0	35287.2372984	0
0.016566	35275.3616323	0	35275.3616323	0
0.018072	35262.8537187	0	35262.8537187	0
0.019578	35249.8380582	0	35249.8380582	0
0.021084	35236.4451626	0	35236.4451626	0
0.02259	35222.8104623	0	35222.8104623	0
0.024096	35209.0731702	0	35209.0731702	0
0.025602	35195.3751091	0	35195.3751091	0
0.027108	35181.8595134	0	35181.8595134	0
0.028614	35168.6698132	0	35168.6698132	0
0.03012	35155.9484106	0	35155.9484106	0
0.031626	35143.8354594	0	35143.8354594	0
0.033132	35132.4676549	0	35132.4676549	0
0.034638	35121.9770469	0	35121.9770469	0
0.036144	35112.489882	0	35112.489882	0
0.03765	35104.1254878	0	35104.1254878	0
0.039156	35096.9952051	0	35096.9952051	0
0.040662	35091.2013788	0	35091.2013788	0
0.042168	35086.8364146	0	35086.8364146	0
0.043674	35083.9819107	0	35083.9819107	0
0.04518	35082.7078696	0	35082.7078696	0
0.046686	35083.0719989	0	35083.0719989	0
0.048192	35085.1191062	0	35085.1191062	0
0.049698	35088.8805927	0	35088.8805927	0
0.051204	35094.3740516	0	35094.3740516	0
0.05271	35101.602974	0	35101.602974	0
0.054216	35110.5565662	0	35110.5565662	0
0.055722	35121.209679	0	35121.209679	0
0.057228	35133.5228527	0	35133.5228527	0
0.058734	35147.4424761	0	35147.4424761	0
0.06024	35162.9010598	0	35162.9010598	0

REPORT DOCUMENTATION PAGE

Form Approved
OMB No. 0704-0188

1. AGENCY USE ONLY *(Leave blank)*

2. REPORT DATE
July 1994

3. REPORT TYPE AND DATES COVERED
Final report

4. TITLE AND SUBTITLE
A Model for Assessment of Dynamic Interaction Between Magnetically Levitated Vehicles and Their Supporting Guideways

5. FUNDING NUMBERS
Project E8691R003, Task 5

6. AUTHOR(S)
James C. Ray, Mostafiz R. Chowdhury

7. PERFORMING ORGANIZATION NAME(S) AND ADDRESS(ES)
U.S. Army Engineer Waterways Experiment Station
3909 Halls Ferry Road
Vicksburg, MS 39180-6199

8. PERFORMING ORGANIZATION REPORT NUMBER
Technical Report SL-94-15

9. SPONSORING/MONITORING AGENCY NAME(S) AND ADDRESS(ES)
See reverse.

10. SPONSORING/MONITORING AGENCY REPORT NUMBER

11. SUPPLEMENTARY NOTES
Available from the National Technical Information Service, 5285 Port Royal Road, Springfield, VA 22161.

12a. DISTRIBUTION/AVAILABILITY STATEMENT
Approved for public release; distribution is unlimited.

12b. DISTRIBUTION CODE

13. ABSTRACT *(Maximum 200 words)*

This report describes the development and verification of an analytical model to study the dynamic interaction between a moving maglev-type vehicle and its supporting flexible guideway, referred to as "vehicle/guideway interaction (VGI)." The theoretical basis and computer coding for the model are described in detail. For verification purposes, the model is applied to a very simple case study and compared to a closed-form solution to the same problem. The results compared very closely, indicating that the model provides an accurate VGI analysis tool. To demonstrate the use of the model, it is applied to an actual Maglev system. The results from this analysis also compared very closely to those from similar analyses using different analytical methods.

The VGI model has two distinct applications: it can be used to accurately predict the vehicle ride quality to be expected from a given vehicle and guideway design and to accurately predict the dynamic deflections and stresses experienced throughout the guideway structure as a result of a vehicle passage. Ride quality results are necessary to design a vehicle suspension system and to determine the guideway stiffness required to meet specific ride quality criteria. Dynamic structural analyses are necessary to produce safe, economical, and accurate guideway designs.

14. SUBJECT TERMS
See reverse.

15. NUMBER OF PAGES
73

16. PRICE CODE

17. SECURITY CLASSIFICATION OF REPORT	18. SECURITY CLASSIFICATION OF THIS PAGE	19. SECURITY CLASSIFICATION OF ABSTRACT	20. LIMITATION OF ABSTRACT
UNCLASSIFIED	UNCLASSIFIED		

9. (Concluded).

Headquarters, U.S. Army Corps of Engineers,
Washington, DC 20314-1000
National Maglev Initiative,
Washington, DC 20590
U.S. Army Engineer Division, Huntsville,
Huntsville, AL 35805

14. (Concluded).

Guideway	Structural analysis
Maglev	Structural dynamics
Magnetic levitation	Vehicle/guideway interaction